OPTIMIZATION OF HYBRID EVAPORATIVE COOLING AND AIRCONDITIONING SYSTEMS - AN ECONOMIC APPROACH

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A Thesis Submitted
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for the Degree of

MASTER OF TECHNOLOGY

by

K. SEKAR

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DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY KANPUR

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CERTIFICATE

Certified that this work on 'Optimization of Hybrid Evaporative Cooling and Airconditioning Systems - An Economic Approach' by K. Sekar has been carried out under my supervision and that this has not been submitted elsewhere for a degree.

May, 1986

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-K. SEKAR

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NOMENCLATURE

A	Cross-sectional area of the structures, m^2
a_g	Absorptivity of the glass
a_t	Absorptivity of the structures
C	Costs, Rs/year
c_p	Specific heat capacity, kJ/kg.°C
c_{th}	Thermal capacity of the wall, kJ/kg.°C
d	Declination angle
F_s	Sunlit fraction
F_{ss}	Angle factor between the surface and sky
h	Enthalpy, kJ/kg
h_a	Enthalpy of air, kJ/kg
$h_{i,a}$	Enthalpy of inside room air, kJ/kg
h_i	Inside convection heat transfer coefficient of the wall, $kW/m^2 °C$
h_o	Outside convection heat transfer coefficient of the wall, $kW/m^2 °C$
$h_{i,g}$	Inside convective heat transfer coefficient of the glass, $kJ/m^2 °C$-h
$h_{o,g}$	Outside heat transfer coefficient of the glass, $kJ/m^2 °C$-S
I_{DN}	Incident direct solar radiation, kW/m^2
I_R	Reflected radiation, kW/m^2
I_d	Diffused radiation, kW/m^2
I_t	Total intensity of solar radiation, kW/m^2
K	Thermal conductivity of the material, kW/m°C

L	Life of the machineries, years
L_d	Life of the desert cooler, years
L_s	Life of the airconditioning system, years
l	Latitude angle
m	Mass of the wall, kg
$\dot{m}_{air}$	Mass flow rate of air, kg/s
$\dot{m}_{ref}$	Mass flow rate of the refrigerant, kg/s
P	Power, kW
$\dot{Q}_c$	Design cooling load, kW
$\dot{Q}$	Total heat transfer, kW
q	Heat transfer through walls, kW/m^2
RH	Relative humidity
r_g	Reflectivity of the glass
r_k	Penalty parameter
T_{adp}	Apparatus dew point temperature, °C
T_{db}	Dry bulb temperature, °C
T_{dg}	Dry bulb temperature of the air leaving the desert cooler, °C
$T_{g,i}$	Temperature of the inside surface of the glass, °C
T_h	Condensing temperature, °C
T_i	Inside room temperature, °C
T_l	Evaporator temperature, °C
T_o	Outside air temperature, °C
T_{om}	Mean outside air temperature, °C
T_{sol}	Sol-air temperature, °C
T_{wb}	Wet bulb temperature, °C
$T_{w,o}$	Temperature of outside surface of the wall, °C

t	Time, hours
t_r	Transmittivity
U	Overall heat transfer coefficient, kJ/m^2-h-°C
V_{room}	Volume of the room, m^3
V	Velocity of inside air, m/s
V_w	Design velocity of outside air, km/h
Vol_{air}	Volume of the air, m^3/min
v	Specific volume, m^3/kg
W_{adp}	Specific humidity at apparatus dew point temperature, kg/kg of dry air
W_i	Specific humidity at inside condition, kg/kg of dry air
$\vec{X}$	Design vector
x	Thickness, m
θ	Incidence angle
ϕ	Relative humidity
ϕ_t	Angle of tilt
α	Wall solar azimuth angle
α_h	Hour angle
β	Altitude angle
Ψ	Sun's Zenith angle
γ	Azimuth angle
δ_p	Profile angle
ρ	Density, kg/m^3
λ	Decrement factor

τ	Timelag factor
η	Efficiency
η_c	Compressor efficiency
η_{η}	Efficiency of the desert cooler
η_m	Mechanical efficiency

ABSTRACT

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A generalised computer programme has been developed for the cooling load calculations on hourly basis and the sol-air temperature variation for different orientation of walls. The effect of heat capacity and timelag factor of the structures on the cooling load estimation has also been studied.

The single-stage vapour-compression airconditioning system has been optimised to get the minimum total cost, for refrigerants R-12 and R-22. This is compared with a

hybrid system of evaporative cooling and airconditioning as to what extent, the latter is more economical than the former. The optimum results have been found out for the inside design conditions of 24°C, 65%; 27.5°C, 56% and 30°C, 60% (dry bulb temperature and relative humidity) being obtained on the basis of Fanger's comfort equation [6]. Various combinations of the operating parameters are also studied.

CHAPTER 1

INTRODUCTION

1.1 AIRCONDITIONING SYSTEM:

Airconditioning is constantly absorbing and applying the latest discoveries of science and engineering in an effort to ease man's environmental problems on earth and in space. The term, airconditioning means the act of putting the air in the proper condition for the desired use. This may mean cleanliness, the temperature and humidity of the air. Airconditioning for human comfort is becoming almost a prerequisite for success in the merchandising field in all except the cooler regions of the country. The gain in customer's good-will is reflected by such an increase in business that the airconditioning pays more than for itself. As any other engineering system, the airconditioning system is controlled by various system parameters. To achieve maximum utilisation and effective implementation, an airconditioning system has to be economical. In view of this, the operating parameters of the system are optimised on the basis of cost as well as functional energy constraints whichever is contemplated. To carry out this objective, there are alternatives available, namely modification of an existing system and design of a new system. The present work is

done on the basis of the former.

1.2. SELECTION OF ENVIRONMENTAL ZONES AND OUTDOOR TEMPERATURES:

Eventhough, there are many applications, the primary function of an airconditioner unit is considered to create temperatures artificially which will be comfortable for human beings. Many investigators have done extensive work on temperature zones for human comfort. Fanger [6] elaborated the methods of getting comfort conditions and also suggested a comfort equation after carrying out extensive field studies. In India, Ramamoorthi [15] has analysed the equation for the Indian weather conditions after taking into account of several variables and suggested the comfort conditions, namely inside dry-bulb temperature, relative humidity and velocity of air. Singh and Prakash [14] presented a new ASHRAE comfort chart and recommended a region of design conditions for the comfort environment. The result of the field studies carried out by Malhotra [10] for the conducive atmosphere pertaining to Indian environment has also been taken into consideration. Based on the energy cost, Whitner [18] has got the higher comfort conditions. Several investigators [1] assumed the daily to outdoor temperature variation as a sinusoidal function which results in Fourier series. In India, Ramamoorthi [15] suggested that the airconditioning

system can be substituted by evaporative cooling system, whenever feasible.

The design of an airconditioning plant is mainly dependent on the cooling load, a function of indoor and outdoor temperatures and internal energy release. The indoor temperatures are chosen on the basis of comfort industrial requirement. The size of the building and various capacities of the equipments are also the primary factors influencing the cooling load calculations. So, great care should be taken in proper selection of the equipments to get the total cost as minimum. It has been a common practice to use air-cooled condensing unit for both residential and commercial applications. This dispenses with water piping, cooling water tower and water treatment, thereby eliminating their costs. But the head pressure of the air-cooled system is usually 10% to 20% higher than that of the water cooled system. Hence the air-cooled system is not used in case of large cooling requirement. For smaller units, the cost and conveniences dominate over other factors. Hence, the air-cooled system is considered in the present analysis.

1.2.1 ANALYSIS OF COOLING LOAD CALCULATIONS:

There are number of parameters affecting cooling load calculations, often difficult to define precisely, and always intricately interrelated. Most of these parameters

can be assembled under the classification of space heat gain. This is necessary because different fundamental principles and equations are used to calculate different modes of energy transfer. The major parts of the space heat gain are (a) heat transfer through the structure of the building, (b) heat generated within space by occupants, lights and appliances and (c) ventilation and infiltration of outdoor air . For the evaluation of structural heat gain, there are different techniques available, namely (a) sol-air temperature method, (b) cooling load temperature difference method, (c) finite difference method and (d) finite element method. Among these, sol-air temperature technique is mostly preferred due to its simple algorithm. The cooling load temperature difference method [3] is implemented by using the transfer function method to compute the one-dimensional transient heat flow through the building structure. Kadambi and Hutchinson [9]have described the finite difference method applied for the heat transfer through the walls. Huebner [7] has elaborated the finite element method in the application of heat flow through structures. The thermal capacity and thickness of the structure are interrelated and presented in the graphical form [2]. The relation between the infiltration load and the volume of the conditioned space is explained in [3,4,12]. Woods et al [19] have modified the ventilation standards by giving emphasis on

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energy saving. The heat generation within the conditioned space has been elaborately given in [12] taking into account of metabolic ratings. The procedure for hourly cooling load calculation is elaborated in [3].

1.2.2 METHOD OF OPTIMISATION:

In general, optimisation can be defined as the act of obtaining the best result under the given circumstances. In design, construction and maintenance of any engineering system, engineers have to take many technological decisions. The ultimate goal of all such decisions is to either minimize the effort required or maximise the desired benefit. Since the required effort or desirable benefit in any practical situation can be expressed as a function of certain design variables, optimisation, in particular, may be considered as the process of finding the conditions that results in the maximum or minimum value of the function, known as objective function. It is very rare that a practical design problem to be unconstrained. Considering this fact, if the expression for the objective function and the constraints are fairly simple in terms of design variables, the classical methods of optimisation technique can be used to solve the problem. On the other hand, if the problem involves the objective function and the constraints which are not stated as explicit functions of design variables or which are too complicated to

manipulate, we should go in for some specialised numerical methods of optimisation. The basic philosophy of these methods to produce a sequence of improved approximations to the optimum value. In the present work, one of such methods called as 'Penalty Function Method' [16] is adopted to obtain the optimum design variables of the airconditioning system for the minimum total cost. The application of this iterative procedure is made possible due to the availability of the computer. These methods are clearly elaborated in [16] .

1.3 PRESENT STUDY:

In the present study, a generalised computer programme has been developed for the accurate evaluation of the cooling load considering daily outdoor temperature variation based on actual as well as approximate. In this regard, the effect of timelag factor and heat capacity of the structures on the structural load has also been found out. The cooling loads have been calculated on the basis of actual hourly daily temperature variations and the sinusoidal temperature variation based on maximum and minimum temperatures. The multiplier has been developed to facilitate the utilisation of the suggested sinusoidal temperature variation; because these data are available for most of Indian cities [11].

For the iterative optimisation technique, a versatile computer programme has been developed which takes care of various factors involved in the total cost of a vapour compression system. Even though there are large number of operating variables present in an air conditioning system, the most important factors have been selected as design variables, giving emphasis on the feasibility of control. For these variables, the constraints are formulated on the basis of practical consideration. A comparative study has also been carried out between the airconditioning systems operating with and without evaporative cooling, for the duration, April to June, in terms of total cost. This aspect is studied for both R-12 and R-22 refrigerants.

CHAPTER 2

COOLING LOAD ESTIMATION

2.1 INTRODUCTION:

The design of an airconditioning plant for a building depends on the use of the building. The cooling load calculation needs the size of the building, its structural details and orientation. In addition, the following quantities are considered:

i) Design conditions: Indoor and outdoor conditions,

ii) Instantaneous heat load: Sensible and latent.

For the given inside design condition, the cooling loads are calculated as follows:

Sensible cooling load: It comprises the heat transfer through structure including solar heat gain. The sensible heat part of the infiltration and ventilation is calculated from the known amount of the air and subsequent difference between enthalpies of the inside and outside air. The sensible heat part of the heat release from the occupants is calculated from the standard data available giving heat release from human beings, ratings of fans, lights, etc.

The latent heat part of the cooling load has been calculated from the infiltration and ventilation air, latent heat release from the occupants inside the room.

2.2 SELECTION OF INDOOR CONDITIONS:

The variation in the weather, building occupancy, and other factors, affecting the load, necessitate carefully coordinated controls to regulate simultaneously the components and equipments to maintain the desired inside conditions. The design indoor conditions play a key role in the design of any comfort airconditioning system. The field studies [10], reveal that in India, for hot humid climates, the comfort zone is between 22°C and 24.5°C, effective temperatures. The analysis [15] carried out on the basis of Fanger's comfort equation recommends the comfort conditions for Indian people as $T_{db} = 24°C$, $RH = 0.65$, $V = 0.8$ m/min; $T_{db} = 27.5°C$, $RH = 0.56$, $V = 1.1$ m/s, and $T_{db} = 30°C$, $RH = 0.60$, $V = 0.7$ m/s. Since, these results lie in the comfort zones recommended in [5,6,14], in the present analysis the same combinations are taken into consideration.

2.3 HOURLY COOLING LOAD:

Most of the factors of the cooling load vary in magnitude over a wide range during a twenty four hour period. As the cyclic changes in the load components are not usually in phase with each other, the analysis is required to establish the resultant maximum cooling load for a building.

2.4 HOURLY VARIATION OF OUTDOOR CONDITIONS:

Hourly variation in the outdoor conditions is available for the limited cities. But the cooling load depends greatly on the ambient temperature variations as compared to that obtained on the basis of design conditions. So a mathematic model [1] has been adopted which gives a close approximation to an average value equal to the mean daily temperature plus an oscillating component with an amplitude equal to one half the mean daily range. For most practical applications, this can further be approximated by a constant with a superimposed sine wave having a 24-hour period. So, the final form of the expression comes out to be a harmonic series of first order:

$$T_o = A + B \text{ Cos } (15\, t - C) \tag{2.1}$$

where T_o = outside air temperature at any time, °C

t = time in hours

A, B, C = the constants being obtained by applying the boundary conditions,

$$\frac{dT_o}{dt} = 0 \text{ when } \begin{cases} T_o = T_{max} \\ T_o = T_{min} \end{cases} \tag{2.2}$$

T_{min} occurs at (Sunrise-1) hrs; T_{max} occurs 12 hrs thereafter. So, the above model is solved for the outdoor air temperature variation by knowing the minimum and maximum temperatures with their corresponding timings, in a day. The result of

this equation has also been verified with actual variation of the temperature and the comparative study has been done in a graphical form, Fig.4.1, for Kanpur. For this purpose, the data for the actual variation of the temperature and relative humidity for the period 1982, 1983 and 1985 have been procured [8] and given in tabular form in Appendix-A.

2.5 SOLAR RADIATION:

The solar radiation is the sole factor for the summer airconditioning system. The total incident solar radiation is given by:

$$I_t = I_{DN} \cos\theta + I_d, \quad k\ W/m^2 \tag{2.3}$$

where the product of I_{DN}, direct normal radiation and the cosine of the incident angle θ, stands for the irradiation on a surface from the sun and I_d refers to the diffused radiation. Simplified general relations for I_{DN} and I_d for a clear sky are given approximately by,

$$I_{DN} = A_1/\exp(B_1/\sin\beta), \quad k\ W/m^2 \qquad 2.4(a)$$

$$I_d = C_1 \cdot I_{DN} \cdot F_{ss}, \quad k\ W/m^2 \qquad 2.4(b)$$

where,

A_1 = apparant solar irradiation at an air mass = 0, $k\ W/m^2$

B_1 = atmospheric extinction coefficient,

C_1 = diffuse radiation factor,

F_{ss} = the angle factor between the surface and the sky,

= 0.5 for vertical surfaces,

= 1.0 for horizontal surfaces,

= $(1 + \mathrm{Cos}\phi_t)/2$ for any other inclined surfaces

and β = the altitude angle.

The values of A_1, B_1 and C_1 of [3] are given in the Appendix-B.

2.5.1 ESTIMATION OF SOL-AIR TEMPERATURE:

For heat transmission calculations, it is convenient to combine the effects of outside air temperature and solar radiation clustered into a single fictitious quantity called sol-air temperature (T_{sol}). The rate of heat transfer, q, from outside to inside surfaces of a sunlit structure may be written as,

$$q = h_o (T_o - T_{w,o}) + a_t \cdot I_t \quad (2.5)$$

where,

h_o = outside air convective heat transfer coefficient, k $W/m^2 - {}^\circ C$

T_o = outside air temperature, °C

$T_{w,o}$ = temperature of the outside surface, °C

a_t = absorptivity of the structure,

and I_t = total intensity oı solar radiation, k W/m^2.

The total intensity of radiation is calculated by the primary angles namely, the latitude (l), hour angle (α_h) and the sun's declination, d. Besides this, there are several other angles as the sun's zenith angle Ψ, altitude angle β, and azimuth angle γ. In Appendix-C, the equations for the estimation of these angles are given.

The equation (2.5) can be modified as

$$q = h_o \, (T_{sol} - T_{w,o}) \tag{2.6}$$

where,

$$T_{sol} = T_o + a_t . I_t / h_o$$

The outside air convection heat transfer coefficient, h_o[9] is calculated as follows:

$$h_o = 14.278+3.559.V_w \quad \text{for very smooth surfaces,} \tag{2.7(a)}$$

$$= 18.423+3.81.V_w \quad \text{for smooth surfaces,} \tag{2.7(b)}$$

$$= 26.55+5.0663.V_w \quad \text{for rough surfaces,} \tag{2.7(c)}$$

$$= 28.64+6.364.V_w \quad \text{for very rough surfaces,} \tag{2.7(d)}$$

where V_w is the design velocity of the outside air in Km/h. In the present study, the variations in the sol-air temperature for a day, on different orientation of the

surfaces presented in a graphical form, Fig.4.2.

2.6 HEAT TRANSFER THROUGH STRUCTURES:

A fraction of the total heat energy is absorbed by the structural material and there is also a timelag for the heat transfer from the outside surface to the inside surface of the structure. The thermal capacity of the walls and ceilings, c_{th}, is taken as,

$$c_{th} = m.\, c_p = \rho\, .\, A\, .\, x.\, c_p \tag{2.8}$$

where m, c_p, ρ, A and x are mass, specific heat, density, cross-sectional area and thickness of the structure, respectively.

Taking into account of timelag and decrement factor,[2] the actual heat transfer through the structure at any time 't' is given by

$$\dot{Q}_t = \sum_{j=1}^{5} [\, U_j\, A_j\, (T_{om} - T_i) + U_j A_j . \lambda_j (T_{o(t-\tau)} - T_{om})\,] \tag{2.9}$$

where T_{om}, λ and τ are the mean sol-air temperature, decrement factor and timelag factor, respectively. In the above expression, U, A, and $T_{o(t-\tau)}$ refer to the overall heat transfer coefficient, cross sectional area and temperature at time $(t-\tau)$, respectively. Here summation symbol stands for four walls and the ceiling.

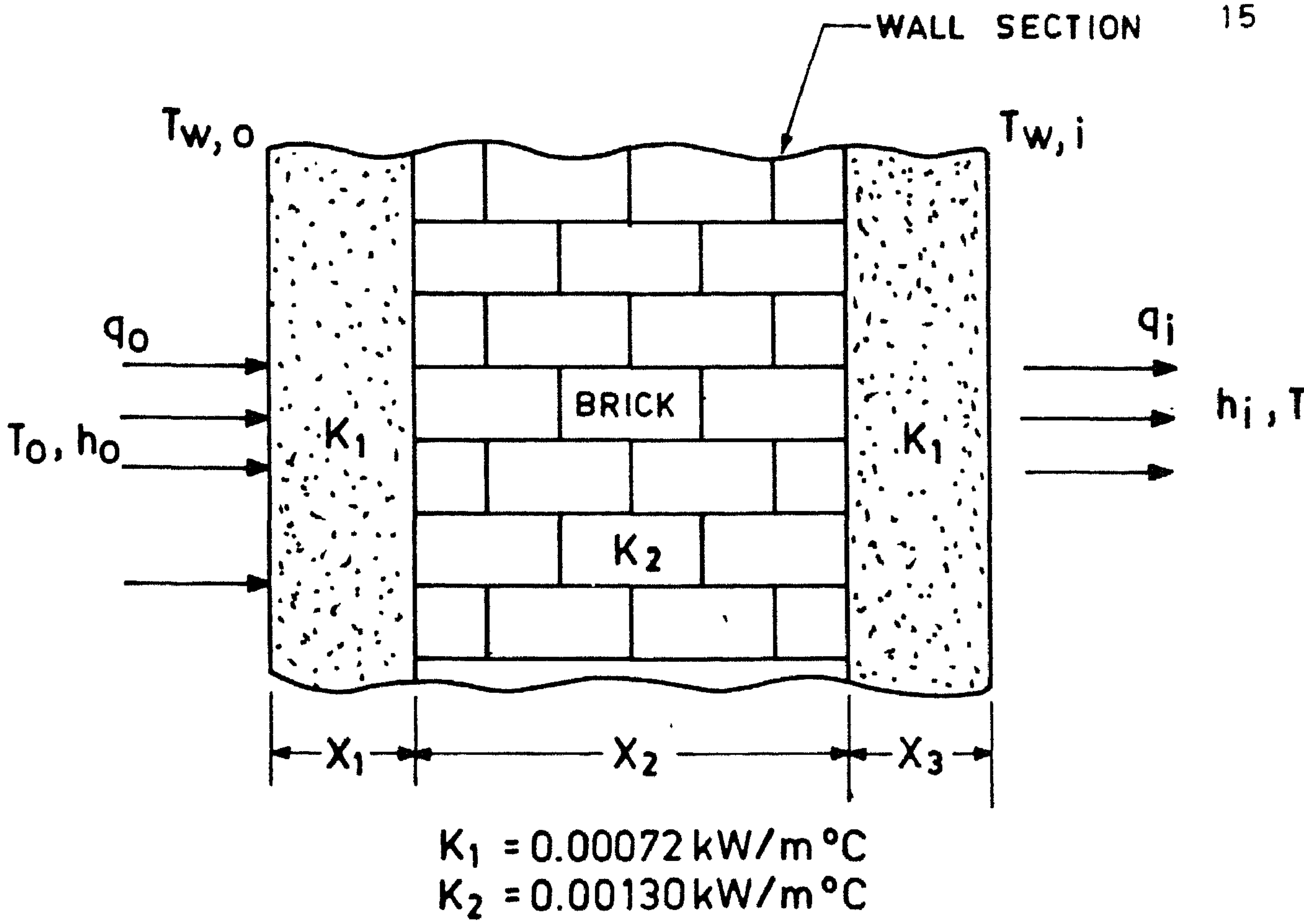


Fig. 2.1 (a) Structural details of the wall.

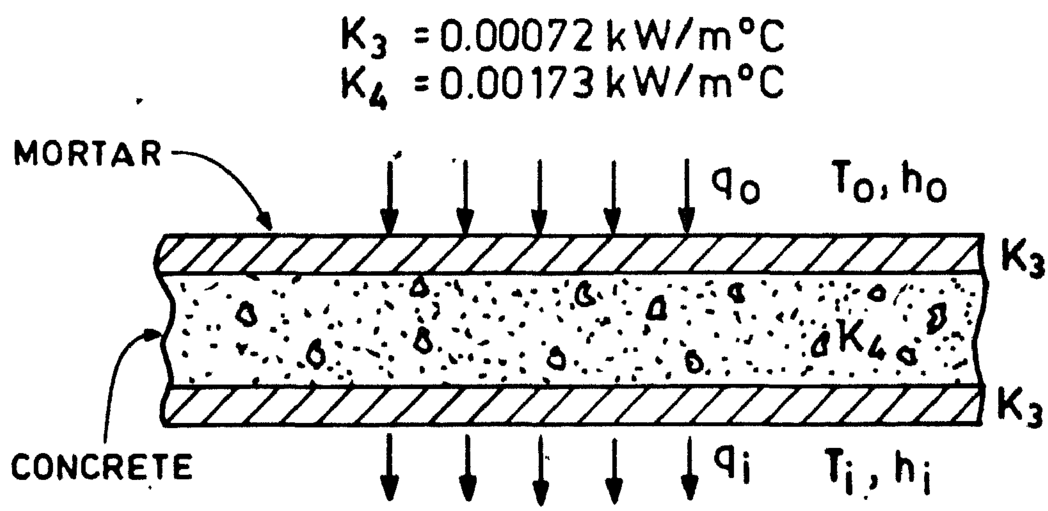


Fig. 2.1(b) Structural details of the ceiling.

Referring to the Fig.2.1, the heat conduction through structure can be written as,

$$q_{st} = h_o (T_o - T_{w,o}) = \frac{(T_{w,o} - T_{w,i})}{\sum_{i=1}^{i=3} (x_i/K_i)} = h_i(T_{w,i} - T_i) \qquad (2.10$$

The above equation can also be written as,

$$q_{st} = U \cdot (T_o - T_i) \qquad (2.11$$

with

$$U = \frac{1}{\left(\frac{1}{h_o} + \sum_{i=1}^{i=3} \frac{x_i}{K_i} + \frac{1}{h_i}\right)}$$

T_o = outside air temperature, °C

T_i = indoor temperature, °C

x_i and K_i = thickness and thermal conductivity of the 'i' th layer of the structural material

h_i = inside convective heat transfer coefficient, $kJ/m^2 - h - °C$

U = overall heat transfer coefficient, kJ/m^2-h-°C.

In the present study, the values of K_i's are taken from [3]. The inside convective heat transfer coefficient is found to be function of temperature difference ($\Delta T = T_{w,i} - T_i$).

For walls;

$$h_i = 4.25\ (\Delta\ T)^{1/4},\ kJ/h - m^2 - {}^\circ C \qquad 2.12($$

For ceiling:

$$h_i = 3.48\ (\Delta T)^{1/4}\ ,\ kJ/h - m^2 - {}^\circ C \qquad 2.12($$

The Newton-Raphson iterative procedure has been used to get the values of 'h_i'. Equation 2.9 is used to calculate the heat transfer through the structure: after considering timelag and decrement factors [2]. The effect of these factors on the cooling loads has been discussed, in Chapter 4.

2.7 HEAT TRANSMISSION THROUGH GLASS WINDOWS:

The incident radiation may be reflected, partly absorbed and the remainder transmitted through the glass materials. Thus, in general, it may be written as:

$$r_g + a_g + t_r = 1 \qquad (2.13$$

where r_g, a_g and t_r are the reflectivity, absorptivity and transmittivity, respectively. The equations for calculating these thermal properties for the glass windows are given in Appendix-D.

From the basic concept of heat transfer, the rate of heat gain through the glass is given by,

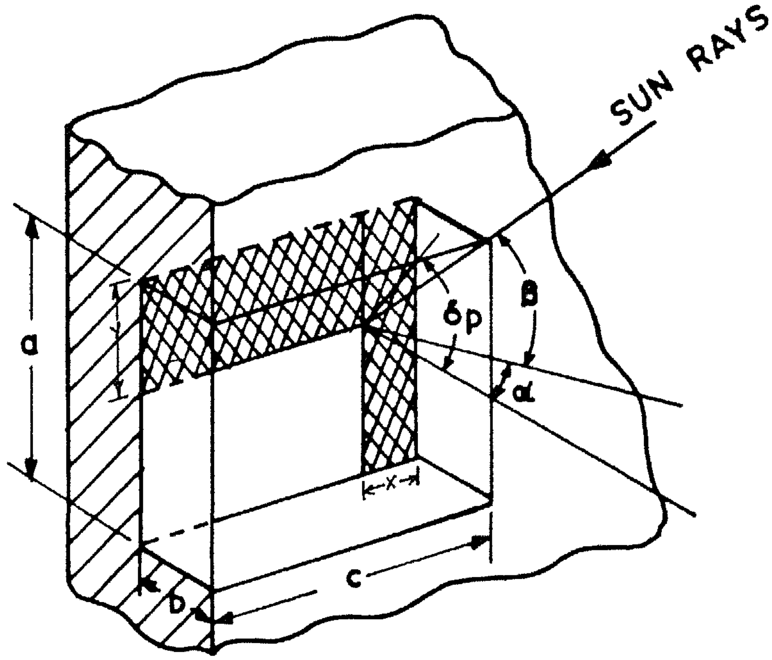


Fig. 2.2(a) Shading of window for a building.

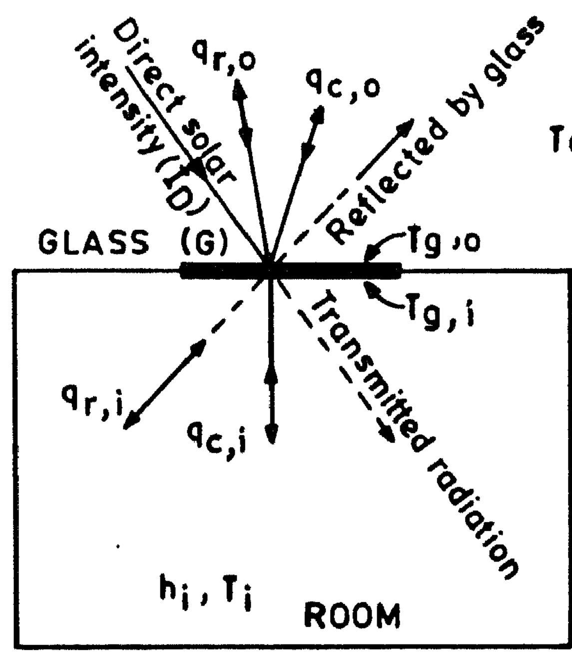


Fig. 2.2(b) Heat transfer through the glass window

$$q_{glass} = F_s \cdot t_{rD} \cdot I_D + t_{rd} \cdot I_d + t_{rR} \cdot I_R + h_{i,g}(T_{g,i} - T_i) \qquad (2.14)$$

where F_s is the sunlit fraction of the window surface, t_{rD}, t_{rd}, and t_{rR} are the transmittivities of the direct, diffused and reflected radiations, respectively. I_D, I_d, and I_R are the intensities of direct, diffused and reflected radiation. $T_{g,i}$ and T_i are the temperatures of inside surface of the glass and indoor conditions, respectively.

In reality, a glass window is not fully exposed to direct sunlight, but partially shaded. Referring to Fig.2.2(a), it can be written as :

$$X = b \,.\, \tan \alpha \qquad 2.15(a)$$

$$Y = b.\, \tan \delta_p \qquad 2.15(b)$$

where the profile angle δ_p is related to the sun's altitude angle β and wall solar azimuth angle α by,

$$\tan \delta_p = \frac{\tan\beta}{\cos\alpha} \qquad (2.16)$$

Finally, the value of sunlit fraction, F_s, is given by,

$$F_s = 1 - P_1.\tan \delta_p - P_2.\tan\alpha + P_1.P_2.\tan \delta_p \tan\alpha \qquad (2.17)$$

where,

$P_1 = b/a$ and $P_2 = b/c$.

Referring to Fig.2.2(b), the energy balance for the glass sheet can be written as,

$$F_s \cdot I_D \cdot a_{gD} + I_d \cdot a_{gd} + I_R \cdot a_{gR} = h_{i,g}(T_{g,i} - T_i) + h_{o,g}(T_{g,o} - T_o) \tag{2.18}$$

where $h_{i,g}$ and $h_{o,g}$ are the surface coefficients given by

$$h_{o,g} = 3.96\times10^{-3} + 9.892\times10^{-4} \cdot V_o, \quad kJ/m^2\text{-}°C\text{-}S \qquad 2.19(a)$$

$$h_{i,g} = 0.237 + (V_i/\text{Height})^{0.5}, \quad kJ/m^2\text{-}°C\text{-}h \qquad 2.19(b)$$

where V_o is the outside air velocity in Km/h and V_i is the indoor air velocity in m/h.

Combining the equations (2.14) and (2.18), the heat transfer through windows and doors including solar heat gain is calculated as

$$\dot{Q}_g = \sum_{k=1}^{N} \Big[(F_s \cdot t_{rD} \cdot I_D + t_{rd} \cdot I_d + t_{rR} I_R) + \frac{(F_s \cdot a_{gD} \cdot I_D + a_{gd} \cdot I_d + a_{gR}\, I_R)}{(1 + h_{o,g}/h_{i,g})} + U\,(T_o - T_i) \Big]_k \tag{2.20}$$

where,

$$U = \frac{1}{\left(\frac{1}{h_{i,g}} + \frac{1}{h_{o,g}}\right)}$$

N = number of windows and doors.

The details of windows and doors are given in Appendix-E. The total heat transfer through the structure is given by the sum of equations (2.9) and (2.20) as:

$$\dot{Q}_{st} = \dot{Q}_t + \dot{Q}_g \tag{2.21}$$

2.8 INFILTRATION LOAD:

Infiltration is the leakage of outdoor air into a building through cracks and opening caused by pressure difference across the boundary surfaces. The net exchange of air may lead to both heat and moisture gain for the space. The volume of infiltration is related in terms of the room volume and tabular values are available in [3]. The expression for the infiltration load is:

$$\dot{Q}_{infil} = \frac{V_{room}}{v_{air}} \times N_{ACH} (h_a - h_{i,a})/24, \text{k W} \tag{2.22}$$

where V_{room} is the volume of the room, v_{air}, is the specific volume of air at outside temperature, N_{ACH} is the number of air changes/day, h_a and h_i, are the enthalpies of outside and inside air, respectively.

2.9 VENTILATION LOAD:

The introduction of outside air into the conditioned space is an important factor, which is necessary to maintain oxygen and odour levels as per standard practice. Ventilation

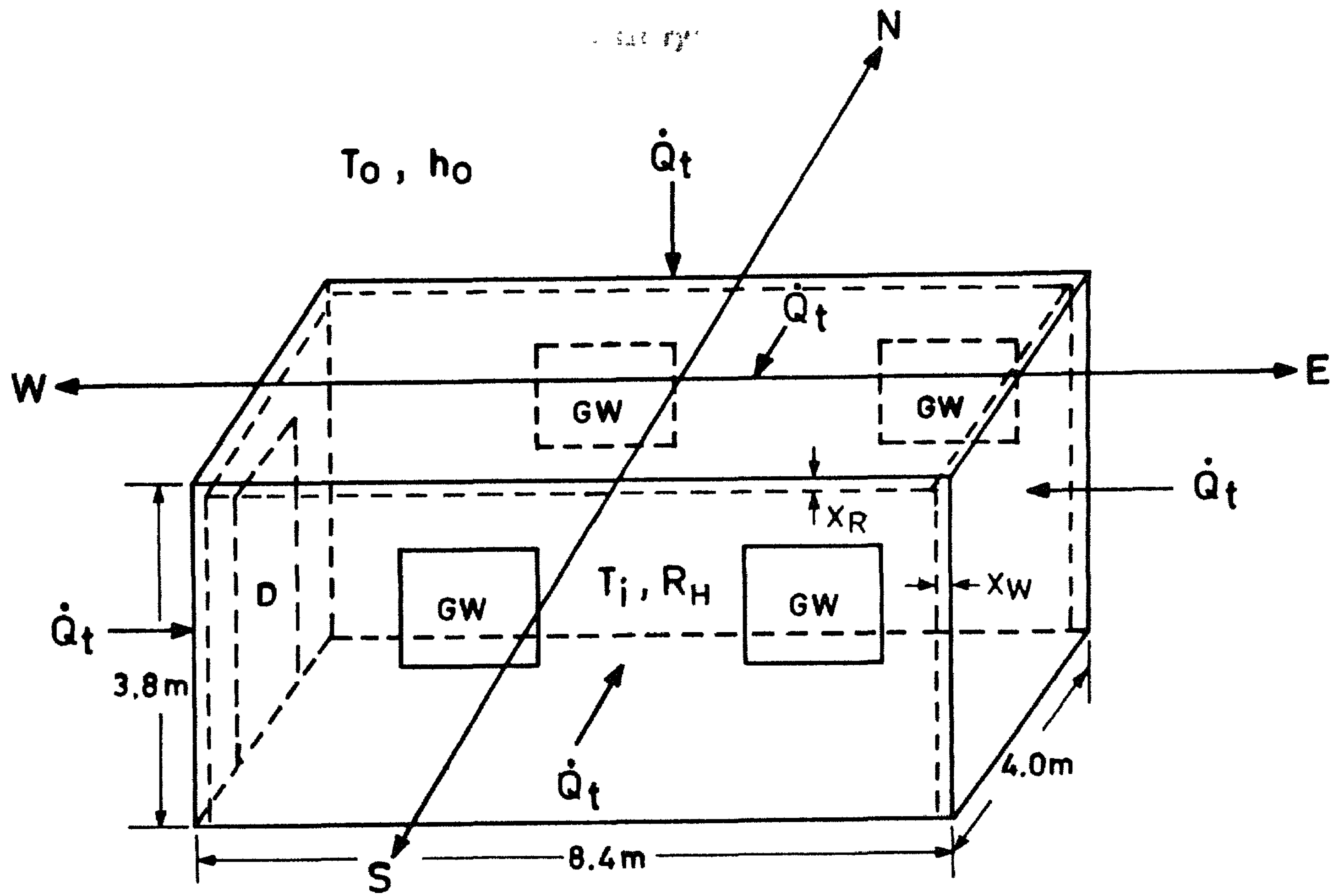
codes and standards recommend a minimum rate of 30 cfm/person. But the recent analysis shows that it is more efficient at 5 cfm/person [19] on the basis of economic consideration. Since the present problem deals with economic aspect, this value has been adopted for the ventilation load calculation, which is given by

$$\dot{Q}_{ventil} = \frac{N_{occupants}}{v_{air}} \times (\text{Ventil/person}) \times (h_a - h_{i,a}) \tag{2.23}$$

where v_{air} is the specific volume of air and h_a, $h_{i,a}$ are the enthalpies of outside and inside air.

2.10 MISCELLANEOUS LOAD:

Apart from transmission, there is a considerable amount of heat energy generated inside the conditioned space from the occupants, lights, electrical equipments and appliances. The thermal energy release from human body at different degree of activities is taken from [12,17]. In the present analysis, the degree of activity is taken as the average of all activities and the corresponding heat release has been calculated. The instant rate of heat gain from lights and other electrical appliances are calculated as,



D - DOOR; GW - GLASS WINDOW; X_W - 0.4m ; X_R - 0.2m

Fig. 2.3 Building model for cooling load estimation.

$$\dot{Q}_{elec} = \text{(Number of electrical appliances x rating of the appliances) + (Number of lights x ratings) x cooling load factor} \tag{2.24}$$

In the present work, the cooling load factor is taken as 0.88 [15] . So the miscellaneous load is calculated as

$$\dot{Q}_{misc} = \dot{Q}_{elec} + \dot{Q}_{occupants} \tag{2.25}$$

The total cooling load is calculated from:

$$\dot{Q}_{total} = \dot{Q}_{st} + \dot{Q}_{infil} + \dot{Q}_{Ventil} + \dot{Q}_{misc} \tag{2.26}$$

Using this procedure, in the present analysis, hourly cooling load calculations have been carried out for a building model, Fig.2.3, for Kanpur. For this estimation, a generalized computer programme has been developed with great care and is given in Appendix-G.

CHAPTER 3

PROBLEM FORMULATION AND OPTIMIZATION

3.1 INTRODUCTION:

The problem formulation for optimization is done on the basis of total cooling load based on the design condition for the equipment selection. The total cost calculated on the basis of fixed and running costs for comfort airconditioning constitutes the objective function in terms of the operating variables having a set of constraints. The optimization problem can be stated as

$$\vec{X} = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_n \end{bmatrix}$$ which maximizes or minimizes $f(\vec{X})$ subject to

the constraints $g_m(\vec{X}) \leq 0$, $m = 1,2,\ldots n_1$, where $\vec{X}$ is an n-dimensional vector called as the design vector, $f(\vec{X})$ is the objective function and $g_m(\vec{X})$ are the constraints.

3.2 DESIGN VARIABLES:

Any engineering system is described by a set of quantities some of which are viewed as variables during the design process. In general, certain quantities are usually fixed at the outset, known as preassigned parameters and all the other quantities are treated as variables in the design process and collectively represented by the vector $\vec{X}$. Even though there are many variables which control

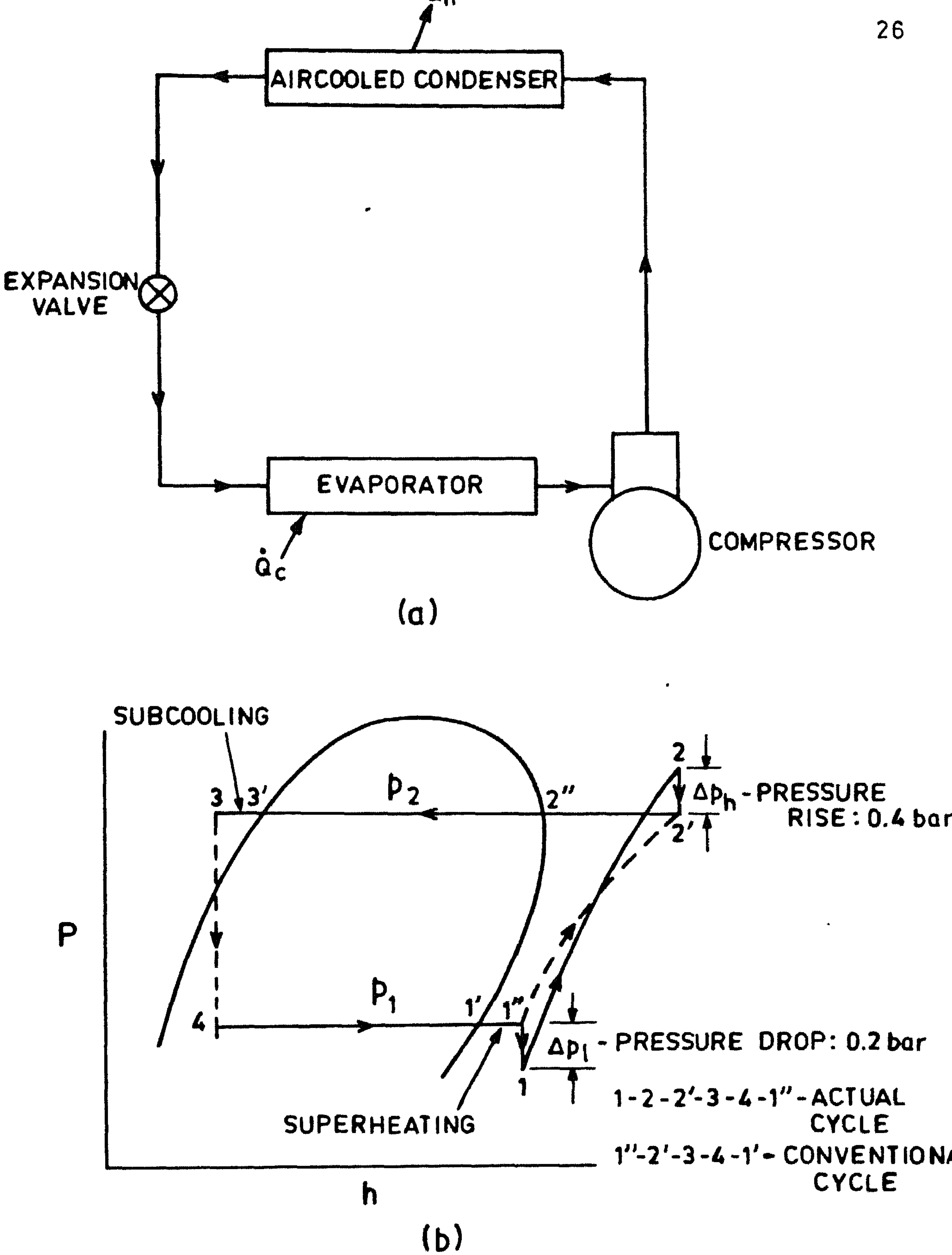


Fig. 3.1 (a) Single-stage vapour-compression system
(b) P-h diagram for single-stage vapour-compression cycle.

the airconditioning performance, inclusion of all these variables will lead to enormous computational effort. Here, an optimal prediction effort has been made to select the variables judiciously.

3.3 SYSTEM DESCRIPTION:

In the present study, a single-stage vapour compression refrigeration system is taken for the analysis having refrigerants R-12 and R-22. The cycle is shown schematically in Fig.3.1. To simulate the cycle close to represent actual processes, a pressure drop in the compressor valves, heat rejection in the compression discharge line and the degrees of subcooling of condensate and superheating of evaporator vapour are considered. The pressure drops in the condenser, evaporator and piping are ignored. In the present analysis, air-cooled condenser has been incorporated in the system.

In the design of the cycle, the pressure drop at the suction line is assumed as 0.2 bar [17]. The increase in pressure at the delivery of the compressor is taken as 0.4 bar [17]. In the present optimization analysis, four variables are considered. They are,

X(1) : Degrees of rise in the condensing temperature, T_h, °C

X(2) : Degrees of fall in evaporator temperature, T_1, °C

X(3) : Degree of superheating, T_s,°C

X(4) : Degree of subcooling, T_c,°C.

3.4 ESTIMATION OF OBJECTIVE FUNCTION:

In the present study, the objective function is taken as the total cost for the comfort airconditioning system, which combines the initial and running costs of the system. If C_I and C_R are the initial and running costs per year respectively, then the total cost/year (C_T) is given by

$$C_T = C_I + C_R, \text{ Rs/year} \tag{3.1}$$

where $$C_I = C_{comp} + C_{evap} + C_{cond} + C_{blower} \quad 3.1(a)$$

C_{comp} = Cost of the compressor, Rs/year

C_{evap} = Cost of the evaporator, Rs/year

C_{cond} = Cost of the condenser, Rs/year

C_{blower} = Cost of the blower , Rs/year

and $$C_R = C_1 + C_2 + C_3 + C_4 \quad 3.1(b)$$

C_1 = Cost of the compressor power, Rs/year

C_2 = Cost of the blower power, Rs/year

C_3 = Cost of the fan power, Rs/year

C_4 = Maintenance cost, Rs/year.

The 'Present Worth Method' is used for the analysis of the economic model.

3.5 OPTIMIZATION TECHNIQUE:

The presence of constraints in a nonlinear programming problem creates more problems while finding the minimum. Several situations can be identified depending on the effect of constraints on the objective function. The simplest situation is when the constraints do not have any influence on the minimum point. From the application point of view on the real problems, there is a possibility of having two or more local minima for the minimisation of the problem. This situation is entirely due to the nature of the objective function contours. By taking into account of all these possibilities, in the present analysis, one of the indirect methods of non-linear programming constrained optimization technique, known as 'Interior Penalty Function' [16] is used.

3.5.1 THE PENALTY FUNCTION APPROACH :

Penalty Function Methods transform the basic optimization problem into alternative formulations such that numerical solutions are sought by solving a sequence of unconstrained minimization problem. For analysis purposes, a basic optimization problem is taken such that $\vec{X}$ minimizes the objective function, $f(\vec{X})$ subject to the constraints $g_m(\vec{X}) \leq 0$, $m = 1,2,\ldots n_1$. This problem is converted into a unconstrained minimization problem by:

$$\phi_k = \phi(\vec{X}, r_k) = f(\vec{X}) + r_k \sum_{j=1}^{n_1} G_j [g(\vec{X})] \qquad (3.2)$$

where G_j is the function the constraints, $g(\vec{X})$, and r_k is the positive constant known as the 'Penalty Parameter'. The function, $\phi(\vec{X}, r_k)$, is optimised for a decreasing sequence values of the penalty parameter, r_k. Evidently, if the 'r_k' is very small, the optimization of $\phi(\vec{X}, r_k)$ amounts to optimization of the objective function, $f(\vec{X})$.

3.5.2 INTERIOR PENALTY FUNCTION METHOD:

In this method, the function $G_j[g(\vec{X})]$ of the equation (3.2), is substituted by $-\dfrac{1}{g_m(\vec{X})}$. The unconstrained minima of $\phi(\vec{X}, r_k)$, all lie in the feasible region and converge to the solution of $f(\vec{X})$ as r_k is varied in the decreasing sequence. So the function $\phi(\vec{X}, r_k)$ is defined as,

$$\phi(\vec{X}, r_k) = f(\vec{X}) - r_k \sum_{m=1}^{n_1} \frac{1}{g_m(\vec{X})} \qquad (3.3)$$

Here, the values of function ϕ will always be greater than f, since $g_m(\vec{X})$ is negative for all feasible points, $\vec{X}$. Since the unconstrained minimization of $\phi(\vec{X}, r_k)$ is to be achieved by changing the sequence of penalty parameter, r_k, in the decreasing order, its initial value should be chosen such that quick convergence is obtained.

Thus, for any feasible starting point, $\vec{X}_1$, the value of r_1 can be taken as,

$$\frac{0.1\ f(\vec{X}_1)}{-\sum_{m=1}^{n_1} \frac{1}{g_m(\vec{X}_1)}} \leq r_1 \leq \frac{f(\vec{X}_1)}{-\sum_{m=1}^{n_1} \frac{1}{g_m(\vec{X}_1)}} \tag{3.4}$$

Once, the initial value of r_k is chosen, the subsequent values of r_k have to chosen such that

$$r_{k+1} < r_k \qquad 3.5(a)$$

In this regard, for convenience purposes, it is taken as,

$$r_{k+1} = D\ r_k, \quad \text{where } 0 < D < 1 \qquad 3.5(b)$$

where in the present analysis, D has been taken to be 0.1.

In the present analysis, the unconstrained minimization of the penalty function $\phi(\vec{X}, r_k)$ is done by 'Davidon-Fletcher-Powell's Method' [16]. This is the best general purpose optimization technique making use of the derivatives that is currently available. The algorithm for this iterative technique is as follcws:

(a) Start with an initial feasible point $\vec{X}_1$ and a nxn positive definite symmetric matrix, $[H_1]$ which is taken as $[I]$.

(b) Compute the gradient of the function, $\nabla \phi_i$ at the point $\vec{X}_1$ and set $S_i = -[H_i] \cdot \nabla\phi_i$. Normalising this, we obtain, $S_{Ni} = \dfrac{S_i}{\sum_{i=1}^{n}[(\nabla\phi_i)^2]^{1/2}}$, where 'n' is the number of variables.

In the present formulation, central difference scheme has been adopted for gradient evaluation.

(c) Find the optimal step length T_i^* in the direction of S_{Ni} and set $\vec{X}_{i+1} = (\vec{X}_i + T_i^* S_{Ni})$. In the present study, the determination of T_i^* is done by 'Cubic Interpolation Method' [16].

(d) Test the new point $\vec{X}_{i+1}$ for optimality using the condition $\dfrac{(\nabla\phi_{i+1} - \nabla\phi_i)}{\nabla\phi_i} \leq \varepsilon_1$, where ε_1 is very small. In the present case, the termination criteria is $\varepsilon_1 \leq 10^{-3}$. If this criteria is not satisfied [H] is updated as follows:

$$[H_{i+1}] = [H_i]+[M_i]+[N_i]$$

where,

$$M_i = \frac{T_i^* \cdot S_{Ni} \cdot S_{Ni}^T}{(S_{Ni}^T \cdot Q_i)}$$

$$N_i = \frac{-(H_i Q_i)\,(H_i \cdot Q_i)^T}{(Q_i^T \cdot H_i \cdot Q_i)}$$

$$Q_i = \nabla\phi(X_{i+1}) - \nabla\phi(X_i) \,.$$

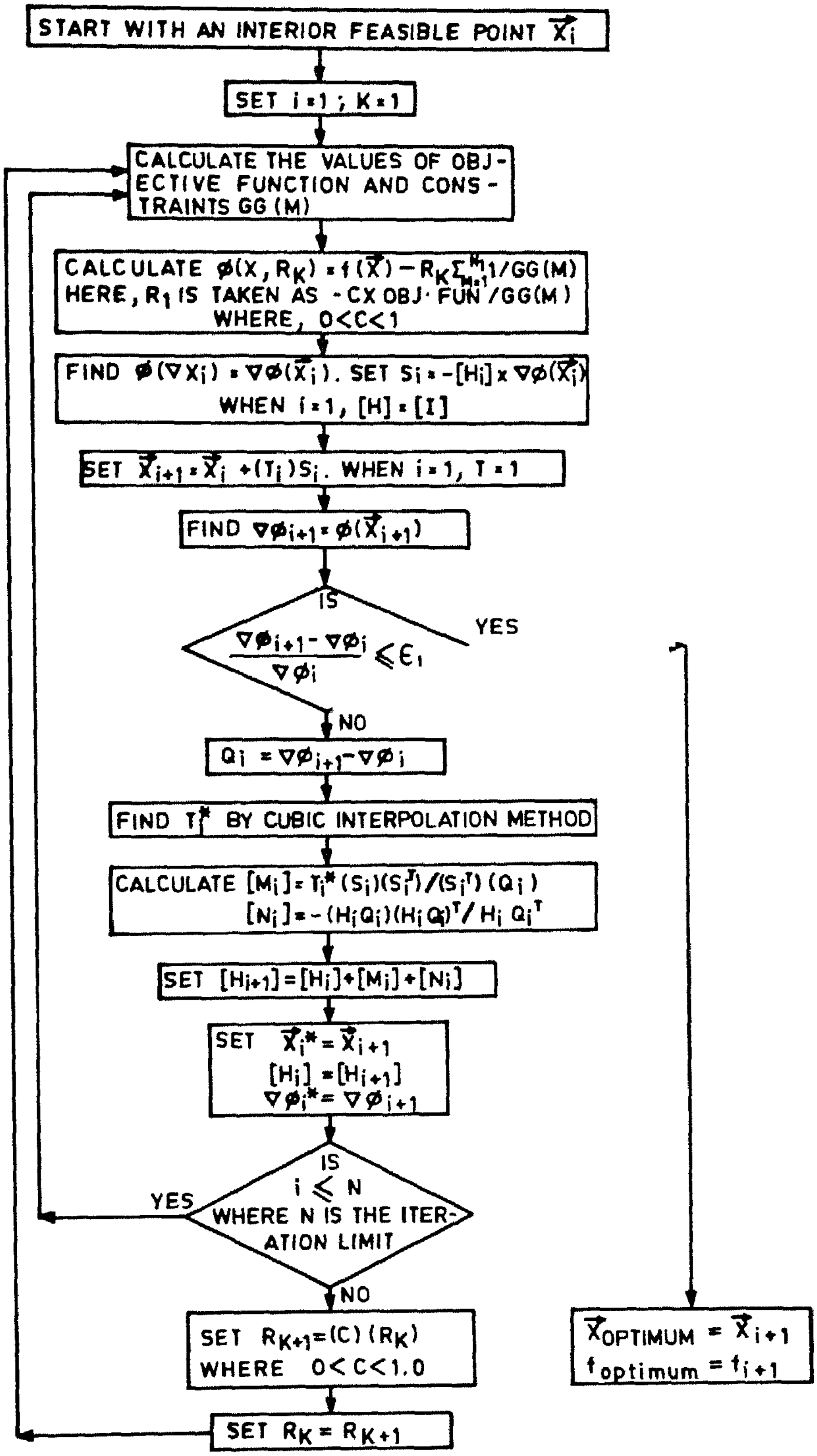


Fig. 3.2 Flow chart for the interior penalty function method using Davidon-Fletcher-Powell's unconstrained optimisation technique.

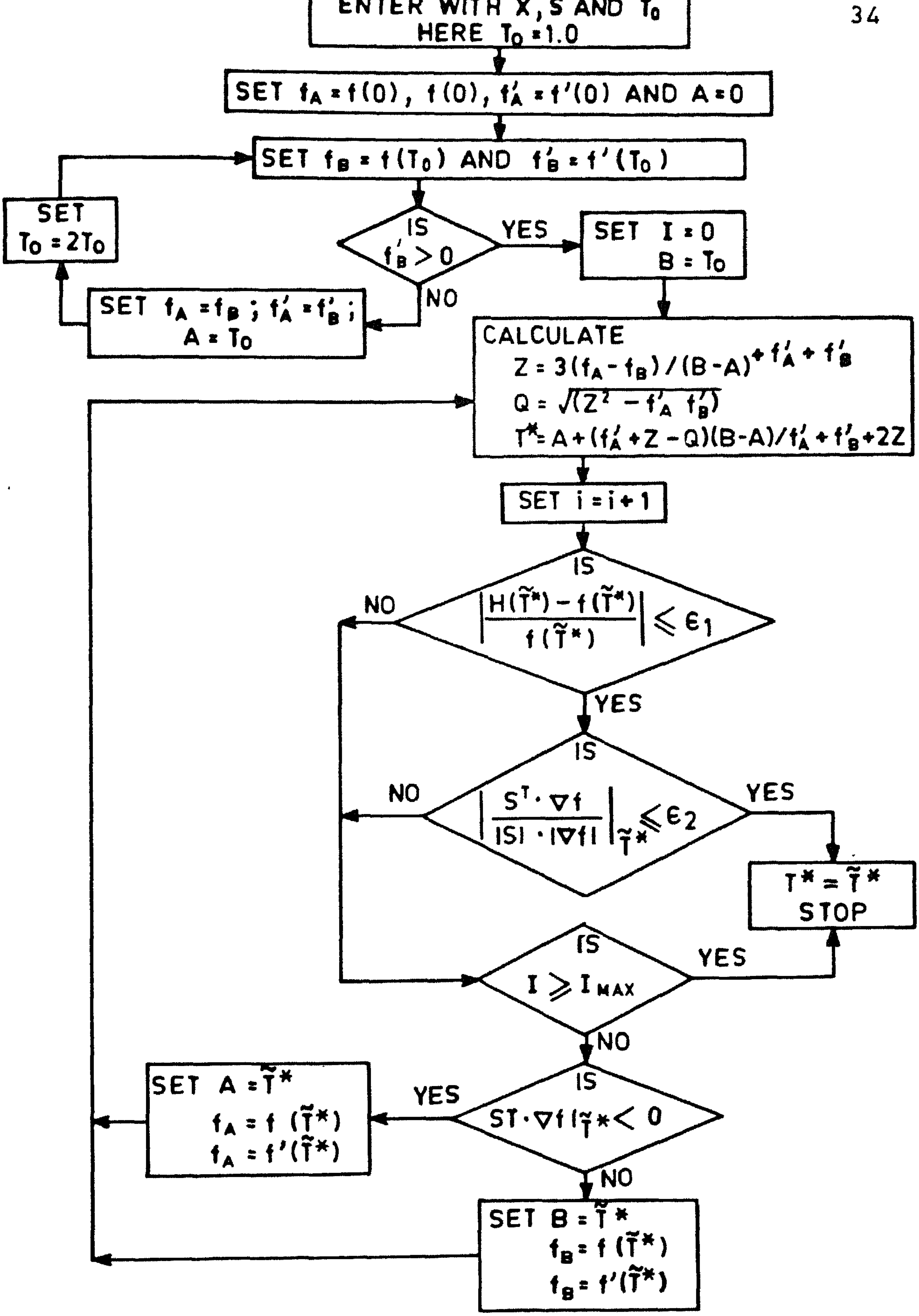


Fig. 3.3 Flow chart for cubic interpolation.

(e) Set the iteration number as i = i+1 and repeat the procedure from the step 'b', until convergence is achieved. Flow charts for 'Interior Penalty Function Method' and 'Cubic Interpolation' are schematically presented in Figs. 3.2 and 3.3, respectively.

3.6 ESTIMATION OF LOADS ON EACH COMPONENT OF THE SYSTEM:

The components of a refrigeration system have to be selected on the basis of the design conditions. The power requirements of the apparatus are evaluated for the design conditions and a factor of safety is multiplied to ensure the capacities of various components from the safety point of view. The condensing and evaporator temperatures of the vapour-compression systems are considered to be the sole factors which influence effectively the design of the airconditioning system. The evaporator temperature is selected on the basis of the indoor conditions whereas the condensing temperature is selected on the basis of design temperature.

So the condenser temperature is taken as,

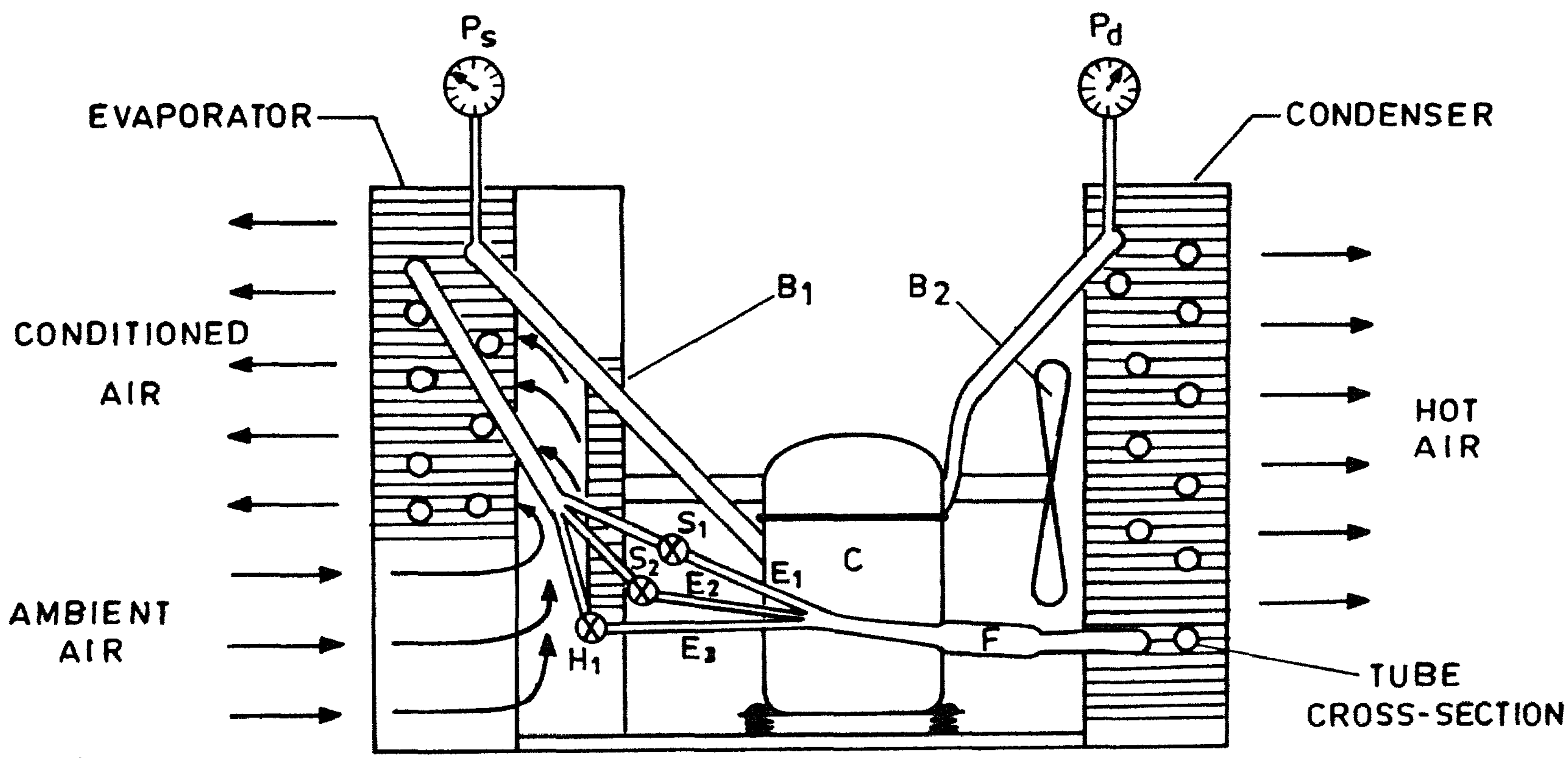
$$T_h = T_o + \Delta T_a + X(1) \tag{3.6}$$

where,

T_o = the design outside air temperature, °C

ΔT_a = the approach temperature of the condenser, °C

$X(1)$ = degrees of rise in condensing temperature, taken as a variable, °C.



B_1 - AXIAL-FLOW FAN
C - COMPRESSOR
E_1, E_2, E_3 - CAPILLARY TUBES
P_d - DISCHARGE PRESSURE GAUGE
H_1 - HOKE VALVE
B_2 - CONDENSER FAN
F - FILTER
S_1, S_2 - SOLENOID VALVES
P_s - SUCTION PRESSURE GAUGE

Fig. 3.4 Experimental set-up of an airconditioner.

The evaporator temperature is chosen on the basis of inside room temperature. So it is taken as,

$$T_1 = T_i - \Delta T_a - X(2) \quad (3.7)$$

where,

T_i = the inside room temperature, °C

ΔT_a = the approach temperature, °C

$X(2)$ = the variation in the lower temperature which is taken as a variable, °C.

In the present analysis, the values of the approach temperature have been taken as 6°C, 10°C and 14°C. To justify these values, experiment has been done on an airconditioner. The pressure gauges were fitted in the suction and discharge line, Fig. 3.4. The airconditioner has been operated and temperatures of air leaving the condenser and evaporator were taken. The air was supplied to both at about 30°C. The average approach temperatures for the condenser and evaporator sides were found to be about 10°C.

The value of T_1 is also checked up in the basis of the apparatus dew point temperature, T_{adp}. Because the lower temperature, T_1, should always be less than T_{adp}. The T_{adp} is calculated in terms of effective sensible heat factor, ESHF, which is given by

$$\text{ESHF} = \frac{1.026\,(T_i - T_{adp})}{1.026\,(T_i - T_{adp}) + 2410\,(W_i - W_{adp})} \tag{3.8}$$

where,

$$\text{ESHF} = \frac{\text{ERSH}}{(\text{ERSH} + \text{ERLH})}$$

$$\text{ERSH} = \text{RSH} + (\text{BPF}) \cdot \text{OASH}$$

$$\text{ERLH} = \text{RLH} + (\text{BPF}) \cdot \text{OALH}$$

W_i = Specific humidity inside the room,kg/kg of dry a

W_{adp} = Specific humidity at apparatus dew point temperature, kg/kg of dry air.

RSH and RLH represent the room sensible and latent heats respectively. OASH and OALH denote the outside air sensible and latent heats, respectively. The effect of the bypass factor (BPF) on the performance of the cooling apparatus is discussed in [4] and 0.15 is the value recommended for row coil. By using Newton-Raphson iterative procedure, the value of T_{adp}, is obtained,for the known values of ESHF.

Referring to the Fig.3.1, after taking the degree of superheat X(3), the temperature at the state point '1' is given by,

$$T_1 = T_1 + X(3) - \Delta T_1 \tag{3.9}$$

where ΔT_1 is the decrease in the lower temperature before compression, °C.

$$\text{Similarly, } T_2 = T_h + \Delta T_h \tag{3.10}$$

where ΔT_h is the increase in the condensing temperature after compression, °C.

The enthalpy values, h of the refrigerants, R-12 and R-22, have been computed from the functional relations of the temperature of the state. These relations are given in the Appendix-F.

$$\text{From the Fig. 3.1, } T_3 = T_3' - X(4) \tag{3.11}$$

where X(4) is the degree of subcooling, °C.

The mass flow rate of the refrigerants, $\dot{m}_{ref}$, is found from :

$$\dot{m}_{ref} = \frac{\dot{Q}_c \times SF}{(h_1 - h_4)} \tag{3.12}$$

where $\dot{Q}_c$ is the cooling load as given in Sec.2.10 and SF is the factor of safety taken as 1.25 [4].

The compressor power is given by:

$$P_{comp} = \frac{\dot{m}_{ref} \; (h_2 - h_1)}{\eta_m}, \text{ kW} \tag{3.13}$$

where η_m is the mechanical efficiency of the compressor taken as 0.85 [12].

For the air-cooled condenser, the load is

$$\dot{Q}_{cond} = \dot{m}_{ref} \; (h_2 - h_3), \text{ kW} \tag{3.14}$$

having $h_2 = h_1 + (\Delta h)_{isentropic} / \eta_c$.

The equation (3.13) can also be equated to the heat gain by the cooling air as

$$\dot{Q}_{cond} = \dot{m}_{air} \times c_{p_{air}} (T_{o,c} - T_{air,i}), \text{kW} \qquad 3.15(a)$$

where,

$\dot{m}_{air}$ mass flow rate of cooling air, kg/s

$c_{p_{air}}$ specific heat of air, kJ/kg.°C

$T_{o,c}$ the outlet temperature of the air,°C

$T_{air,i}$ the inlet temperature of the air,°C.

In the above equation, the temperature differential is also equal to the variation in the condensing temperature (T_h) denoted by X(1). So the equation 3.15(a) can be rewritten as,

$$\dot{Q}_{cond} = \dot{m}_{air} \times c_{p_{air}} \times X(1), \text{ kW} \qquad 3.15(b)$$

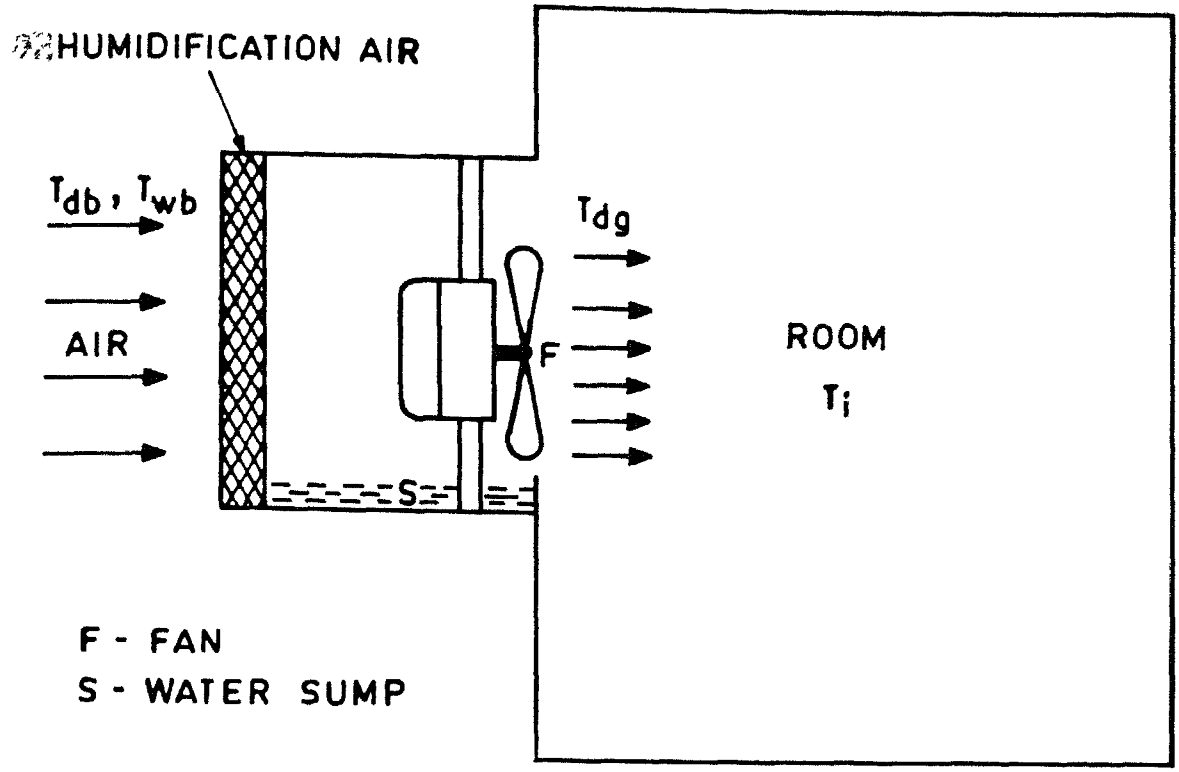
By combining equations 3.14 and 3.15b,

$$\dot{m}_{air} = \frac{\dot{m}_{ref}(h_2 - h_3)}{c_{p_{air}} \times X(1)}, \text{ kg/s} \qquad (3.16)$$

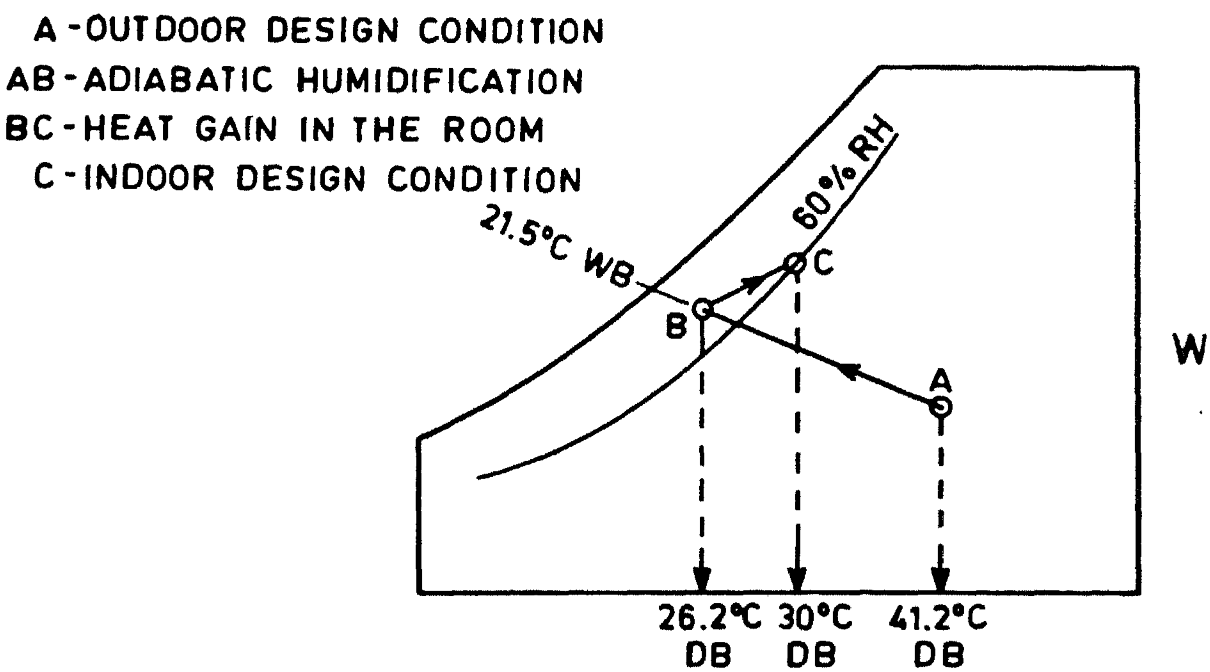
From equation (3.16), the volume of the air handled by the fan for the condenser is calculated as.

$$Vol_{air} = \frac{\dot{m}_{air} \times 60}{\rho_{air}}, \text{ m}^3/\text{min} \qquad (3.17)$$

The fan power is correlated with its capacity (m^3/min) in the form of a quadratic equation. For the purpose, a



(a)



(b)

Fig. 3.5 (a) Evaporative cooling arrangement in room.
(b) Evaporative cooling process.

set of volume flow rate and power consumption readings have been taken for three blowers, experimentally. These readings are plugged in the quadratic equation and they have been solved for the constants. So this procedure finally results in the form:

$$P_{blower} = \frac{3.429\,(Vol_{air})^2 + 1.143\,(Vol_{air}) + 37.67}{1000 \times \eta_m}, \text{ kW} \tag{3.18}$$

where,

Vol_{air} = The volume flow rate of air, m^3/s

η_m = The mechanical efficiency of the fan (0.85)

3.7 EVAPORATIVE COOLING PROCESS:

Figure 3.5(a) exhibits the evaporative cooling arrangement for the room. The unsaturated outside air comes in contact with the wetted surface where the evaporation of the water lowers the humidified air temperature. This process is schematically represented by process AB in 3.5 (b). The temperature of the humidified air is related to the desert cooler efficiency (η_D) as

$$T_{dg} = T_{db} - \eta_D\,(T_{db} - T_{wb}) \tag{3.19}$$

where,

T_{db} = dry bulb temperature of the outside air, °C

T_{wb} = wet bulb temperature of the outside air, °C

T_{dg} = dry bulb temperature of the air leaving the desert cooler, °C .

The volume of humidified air for the airconditioning is calculated from:

$$V_{d,air} = \frac{\dot{Q}_c}{\rho_{air}\,(h_{a,o} - h_{a,i})}$$

$$= \frac{\dot{Q}_c}{\rho_{air} \cdot 1.026 \cdot (T_i - T_{dg})} \qquad (3.20)$$

where ρ_{air} is the density of the humid air.

3.8 CONSTRAINTS FOR THE PROBLEM:

As mentioned earlier, four variables, being restricted between some practical values, are taken into consideration for the optimization. So, these constraints are mathematically expressed as follows:

$$4.0 \leq X(1) \leq 15.0 \qquad 3.21(a)$$

$$2.0 \leq X(2) \leq 5.0 \qquad 3.21(b)$$

$$1.0 \leq X(3) \leq 4.0 \qquad 3.21(c)$$

$$1.0 \leq X(4) \leq 4.0 \qquad 3.21(d)$$

The constraints are transformed into equality constraints by taking the lower and upper limits, separately. As a result, they are transformed in the following form:

$$GG(1) = 4.0-X(1) \qquad 3.22(a)$$

$$GG(2) = X(1) - 15.0 \qquad 3.22(b)$$

$$GG(3) = 2.0 - X(2) \qquad 3.22(c)$$

$$GG(4) = X(2) - 5.0 \qquad 3.22(d)$$

$$GG(5) = 1.0 - X(3) \tag{3.22(e)}$$

$$GG(6) = X(3) - 4.0 \tag{3.22(f)}$$

$$GG(7) = 1.0 - X(4) \tag{3.22(g)}$$

$$GG(8) = X(4) - 4.0 \tag{3.22(h)}$$

After normalising the equations, the constraints are considered for the optimization of the problem.

3.9 ESTIMATION OF INITIAL COST:

As mentioned in Sec. 3.6, the design cooling load is calculated for Kanpur, in May as in [9] . After evaluating these loads, the equipments are selected on the basis of their capacities. In this regard, a due consideration has been given to purchase the equipments on the basis of recommended capacity ranges available in the market. A stepwise variation is adopted for the costs and capacities of the various equipments. This feature is very well incorporated in the computer programme.

Thus, the total initial cost for the system is expressed as

$$CI = (C_{comp} + C_{evap} + C_{cond} + C_{blower}) \times MF/L_s \tag{3.23}$$

where,

C_{comp} = cost of the compressor, Rs

C_{evap} = cost of the evaporator, Rs

C_{cond} = cost of the condenser, Rs

C_{blower} = cost of the blower, Rs

MF = Multiplication factor to take care of the miscellaneous cost (1.2)

L_s = life of the system, years.

When a hybrid system of airconditioning and desert cooler is adopted between the months April and June, if feasible, the initial cost, C_I', is considered to be higher and given by,

$$C_I' = C_I + C_{Desert}/L_d \tag{3.24}$$

where,

C_I = initial cost of the airconditioning system, Rs/year

C_{Desert} = cost of the desert cooler, Rs

L_d = life of the desert cooler , years.

In the present work, both these cases are studied separately and results are compared.

3.10 ESTIMATION OF RUNNING COST:

For the running cost, the electricity cost is the primary factor to be considered. Keeping in view, the variation in electricity cost, a futuristic statistical approach has been adopted to make the whole formulation as a generalised algorithm [13]. Proceeding as in [13], the electric charge, (C_e), is found to be:

$$C_e = \frac{1.293}{0.97986 + \exp(-0.09338\ I)} \qquad (3.25)$$

where, I refers to the duration of the years from the present one.

The effective electricity cost is given by [15]:

$$C_E = \frac{1}{L} \sum_{i=1}^{L} \frac{C_e}{(1+R)^{i-1}} \qquad (3.26)$$

where,

C_e = cost of electricity in the ith year, Rs/kWh

R = interest rate

L = life of the machineries, years

So, the running cost for compressor (C_1) is given by,

$$C_1 = P_{comp} \times C_E,\ \text{Rs/h} \qquad (3.27)$$

For the evaporator fan,

$$C_2 = P_{fan} \times C_E,\ \text{Rs/h} \qquad (3.28)$$

For the condenser fan,

$$C_3 = P_{fan} \times C_E,\ \text{Rs/h} \qquad (3.29)$$

The running cost of all these accessories is calculated on the hourly basis by taking into account by sinusoidal variation of outside air temperature and summed upto get the running cost/year. The maintenance cost is taken to be 10% of the initial investment [15].

Mathematically, it can be expressed as*,

$$C_4 = (0.1)\ C_I \quad \frac{(1+R)^L - 1}{R(1+R)^L - 1} \tag{3.30}$$

The total running cost is given by:

$$C_R = C_1 + C_2 + C_3 + C_4, \text{ Rs/year} \tag{3.31}$$

In the case of combined operation of airconditioning and desert cooler the running cost includes for the desert cooler only for the feasible period during April to June. Then the total running cost becomes,

$$C_R = C_E \times D.C_{rating} + (C_1+C_2+C_3)^{**} + C_4 \tag{3.32}$$

where C_E the effective electricity cost, Rs/kwh

$D.C_{rating}$ The desert cooler rating, kW.

Thus, the total cost of the system, C_T, is given by

$$C_T = C_R + C_I, \text{ Rs/year} \tag{3.33}$$

---

*In Eq.(3.30) when desert cooler is also used, C_I is replaced by C_I' of Eq.(3.24).

**This operative cost is for the period during which it operates alone.

In the case of hybrid system, it is

$$C_T = C_R + C_I' \text{ , Rs/year} \tag{3.34}$$

For generalisation purpose, the equations 3.33 and 3.34 are modified as,

$$\frac{C_T}{C_E} = \frac{(C_R + C_I)}{C_E} \text{ , kWh/year} \qquad 3.35(a)$$

$$\frac{C_T}{C_E} = \frac{(C_R + C_I')}{C_E} \text{ , kWh/year} \qquad 3.35(b)$$

Thus the ratio C_T/C_E can be written in functional form as, $C_T/C_E = f(X_1, X_2, X_3, X_4)$, where X_i's are the variables taken for the present study to get the optimum values for the minimum of the ratio C_T/C_E.

In the present analysis, all these possible combinations are incorporated in a generalised computer programme for optimization and the same is incorporated in Appendix-G.

CHAPTER 4

RESULTS AND DISCUSSION

This chapter deals with the calculation of cooling loads on the basis of the theoretical and actual temperature variations having the effect of heat capacity and timelag factor of the structures. The economic criterion has been adopted for the optimum choice of the airconditioning systems using R-12 and R-22 refrigerants. A comparison of a hybrid system (evaporative cooling and airconditioning systems) with a conventional airconditioning system is discussed under optimum conditions. For estimation of the optimum cost, C_T/C_E, the life of the airconditioner and the desert cooler are taken as 20 years and 10 years, respectively. The following are the results obtained in the present work.

4.1 THEORETICAL AND ACTUAL TEMPERATURE VARIATION:

Figure 4.1 gives the actual temperature variation with time as well as the suggested sinusoidal variation in temperature based on maximum and minimum temperatures of the day. The actual temperature is slightly higher from 8 A.M. to 3 P.M. than the theoretical value before the peak hour. Thereafter, the actual temperature is lower than that of the theoretical value. The areas under both curves reveal small difference (about 4 to 7%). For the airconditioning

TABLE : 4.1(a) : INSIDE CONDITIONS OF 24°C DB AND 65% RH

Year	Outside air temperature variation	Total Running Loads for 10 Hours (In kW)						
		April	May	June	July	August	September	October
1982	Actual	88.66	96.29	86.68	69.09	65.17	64.66	47.28
	Theoretical	84.44	88.72	79.64	63.96	64.26	58.19	44.35
	% difference	4.76	7.86	8.12	7.43	1.39	10.01	6.19
1983	Actual	75.34	85.80	79.51	66.20	62.75	58.29	30.29
	Theoretical	70.95	82.39	76.50	60.94	54.01	57.36	29.23
	% difference	5.83	3.97	3.78	7.95	13.92	1.59	3.49
1985	Actual	59.75	84.21	82.32	69.88	71.32	74.01	41.32
	Theoretical	58.29	82.90	78.91	68.32	69.09	71.86	38.10
	% difference	2.48	1.55	4.14	2.23	3.13	2.88	7.79

TABLE:4.1(b) : FOR INSIDE CONDITIONS OF 27.5°C DB AND 56% RH

Year	Outside air temperature variation	Total Running Loads for 10 Hours (In kW)						
		April	May	June	July	August	September	October
1982	Actual	77.05	82.20	75.89	57.24	52.57	51.68	35.78
	Theoretical	72.33	75.68	66.92	51.13	49.65	55.75	33.25
	% difference	6.13	7.93	11.82	10.67	5.55	11.47	7.07
1983	Actual	68.52	78.35	68.91	59.06	55.09	51.64	28.40
	Theoretical	65.71	74.29	63.90	54.66	50.44	49.32	25.36
	% difference	4.10	5.18	7.42	7.45	8.44	4.49	10.7
1985	Actual	55.78	78.92	74.65	66.89	69.19	71.22	38.57
	Theoretical	53.37	76.01	71.92	64.09	67.72	68.82	35.09
	% difference	4.32	3.68	3.66	4.19	2.17	3.49	9.02

TABLE : 4.1(c) : FOR INSIDE CONDITIONS 30°C DB AND 60% RH

Year	Outside air temperature variation	Total Running Loads for 10 Hours (In kW)						
		April	May	June	July	August	September	October
1982	Actual	68.68	73.71	67.68	48.65	43.77	41.47	28.36
	Theoretical	63.58	68.91	58.29	42.27	38.05	36.82	24.64
	% difference	7.43	6.51	13.82	13.11	13.07	11.21	13.12
1983	Actual	55.89	65.88	56.31	46.08	40.5	41.88	18.17
	Theoretical	50.18	61.63	55.34	40.52	35.90	37.58	15.71
	% difference	10.08	6.45	1.70	12.06	11.4	10.26	13.54
1985	Actual	51.02	78.43	72.61	67.45	68.33	69.02	38.11
	Theoretical	49.81	73.92	68.88	62.21	64.66	67.11	37.31
	% difference	2.37	5.75	5.17	7.77	5.38	2.84	2.10

system, the working hours have been taken to be 8 A.M. to 5 P.M.*. An underestimation of cooling loads by using sinusoidal variation is found in this. This deviation is calculated for three years, 1982, 1983, 1985 and the same are tabulated in Tables 4.1a to c, for different indoor conditions. From these values, the average deviation is calculated as 6%. So a multiplier 1.06 can be used to get the actual total running loads for 10 hours.

The variation in the sol-air temperatures for different directions is shown in Fig. 4.2. It indicates that the maximum sol-air temperature occurs on the western facing wall at 3 P.M.

4.2 EFFECT OF HEAT CAPACITY AND TIMELAG FACTOR ON THE COOLING LOADS:

Figures 4.3(a) and 4.3(b) represent the effect of the heat capacity and timelag of the structures on the hourly cooling load estimation. From this, it is found out that the estimation of the cooling load without taking into account of heat capacity of the structures leads to an overestimation of 20.29% than that of the cooling load

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* The computer programme is quite general and can be used for any duration.

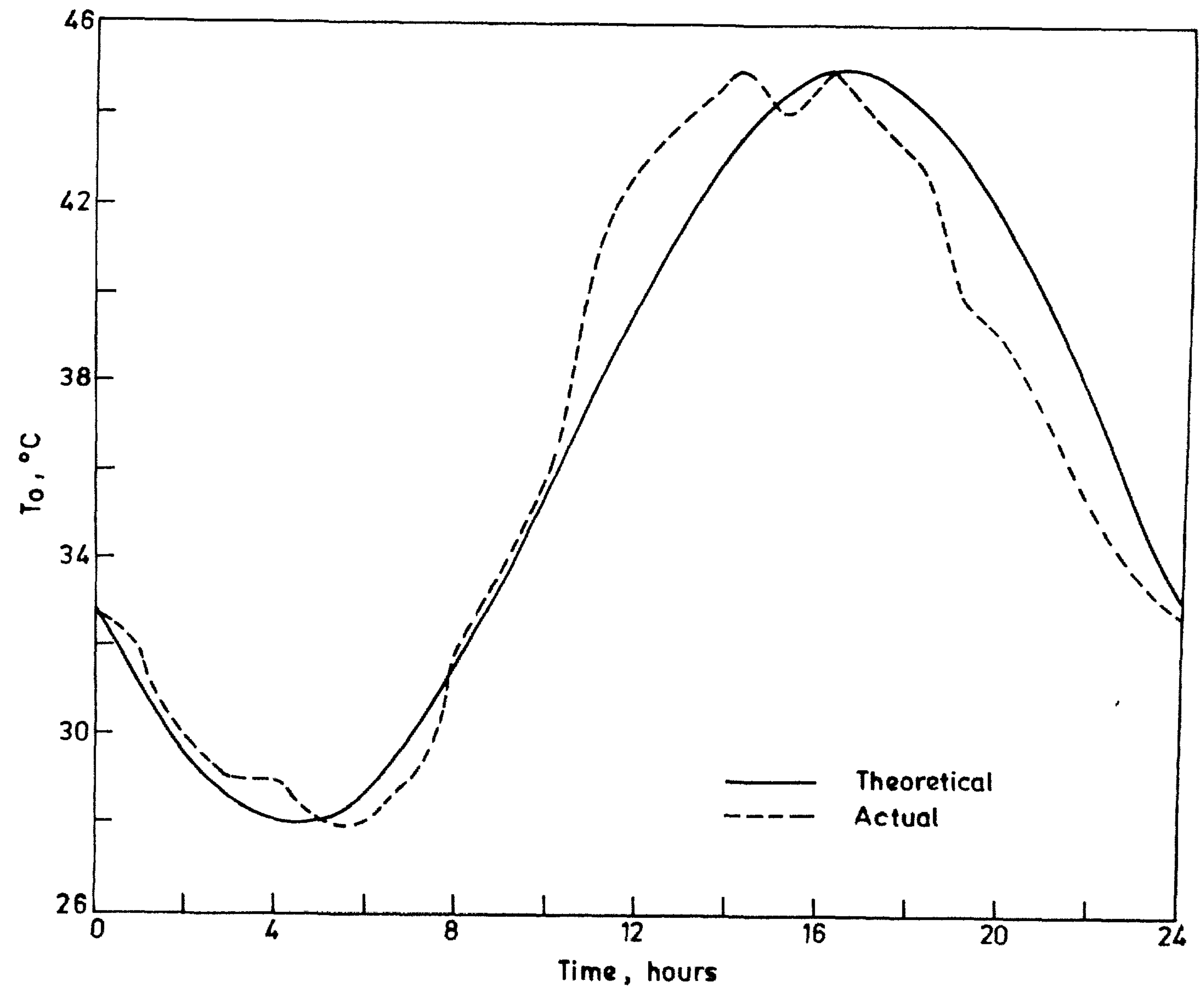

Fig. 4.1 Variation in outside air temperature with respect to time in May at Kanpur.

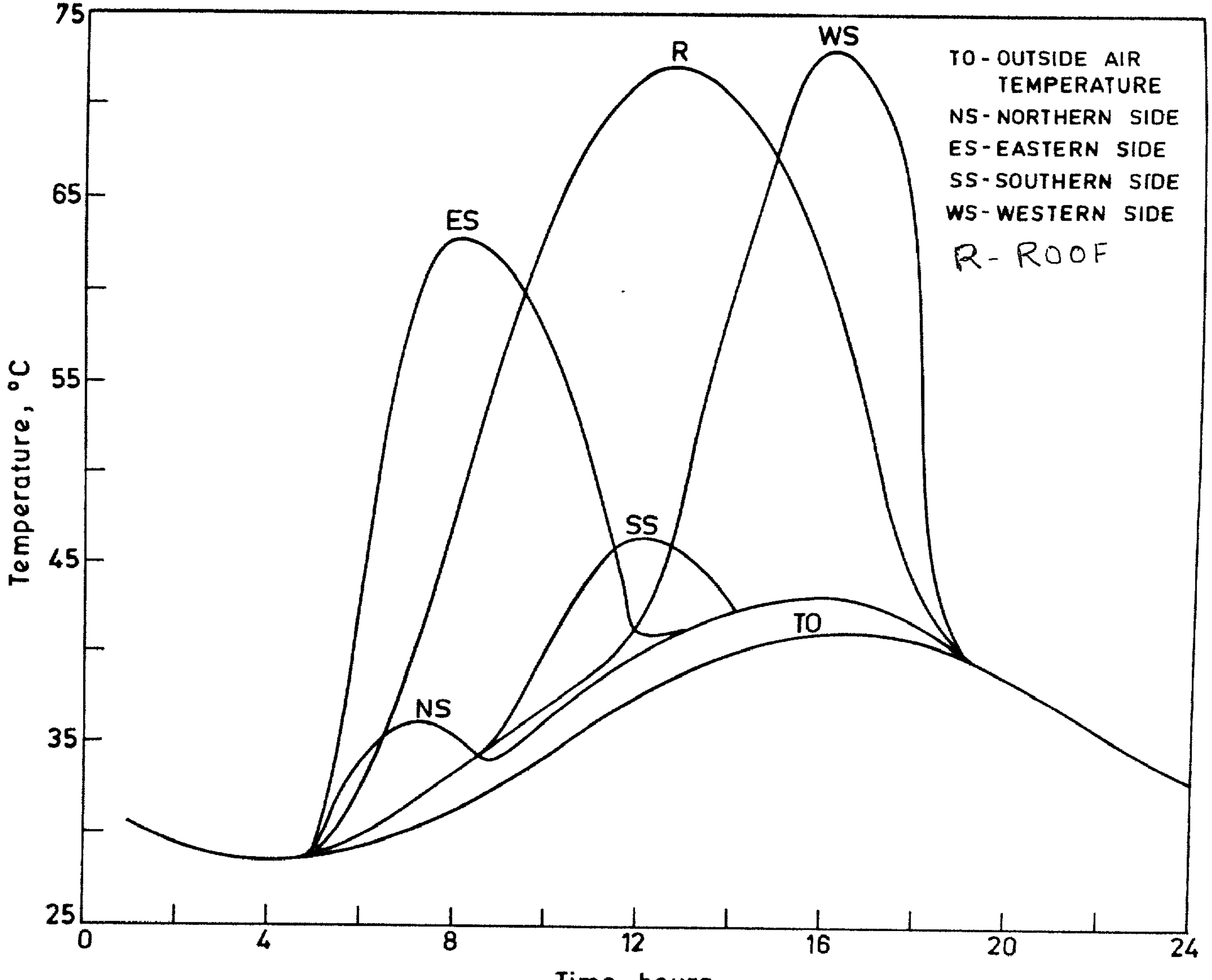

Fig. 4.2 Variation in sol-air temperature and outside air temperature w.r.t. time at Kanpur.

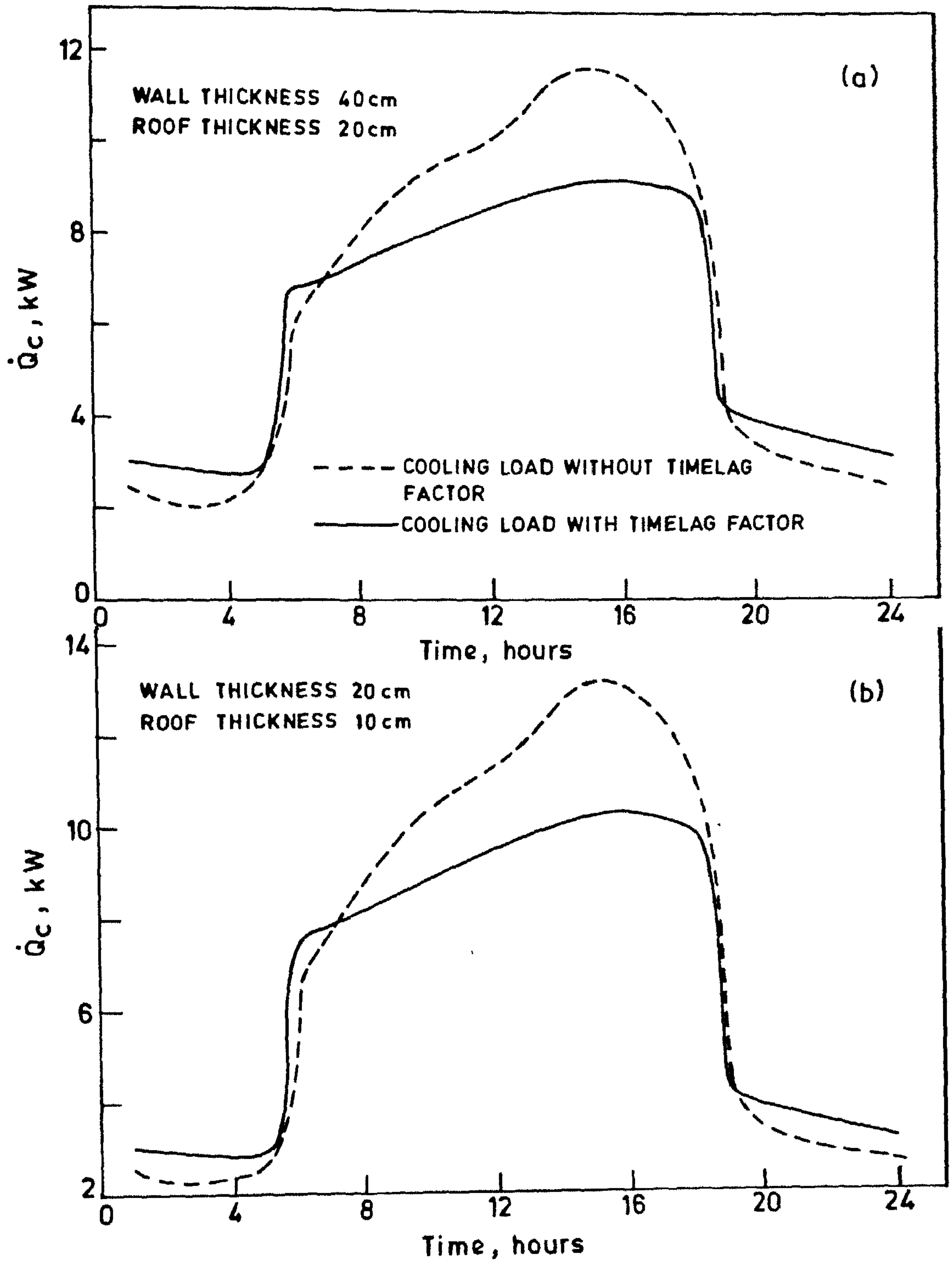

Fig. 4.3 (a) & (b) Variation in cooling load with respect to time.

calculated for the structure with the heat capacity. This difference is quite pronounce. When the wall thickness was halved, the above is found to be 18.02%.

4.3 OPTIMUM RESULTS FOR VARIOUS INDOOR CONDITIONS:

For the indoor conditions of 24°C DB and 65% RH, the design cooling load is 9.8% and 24.5% higher than that for the indoor conditions of 27.5°C DB, 56% RH and 30°C DB, 60% RH, respectively. Figures 4.4 and 4.5 represent the variations in initial, running and total costs with respect to the rise in condensing temperature (T_h) of the system, for R-22 and R-12 refrigerants, respectively. In this case, the approach temperature is taken as 10°C, while the other design variables, the fall in lower temperature ΔT_1, the degree of superheat ΔT_{sup}, and the degree of subcooling ΔT_{sub} are maintained at the optimum values. The optimum total cost, C_T/C_E, for R-12 refrigerant system is 4.752% less than that of R-22 refrigerant system for the same operating variables.

Figure 4.6 shows that the optimum total cost, (C_T/C_E), for a hybrid system of evaporative cooling and R-12 airconditioning system yields 4% gain over the conventional airconditioner used from April to October, every year. This is applicable for the indoor conduction of 27.5°C dry-bulb temperature and 56% RH, with approach temperature as 10°C.

Figure 4.7 depicts the comparison of the optimum total cost between a conventional airconditioning system and the system with evaporative cooling. The optimum total cost for the conventional airconditioning system is 5.562% higher than that the system used with evaporative cooling facility. The variation in the optimum total cost (C_T/C_E) with respect to the fall in the lower temperature of the system, ΔT_l is displayed in Fig.4.8. The effect of ΔT_l on the optimum total cost is less than that of ΔT_h. Figures 4.9(a) and (b) show that the degrees of superheat and subcooling have relatively little effect on the optimum total cost of the system.

Figure 4.10 represents the variation in the optimum total cost with respect to the approach temperature of the systems for various indoor conditions. The approach temperature is **almost** proportional to the optimum total cost or vice versa. The various values of optimum variables for R-12 and R-22 refrigerant systems are given in the Table 4.2, along with the optimum total cost, penalty function values and the penalty parameters.

TABLE : 4.2 : OPTIMUM QUANTITIES FOR R-12 AND R-22 AIRCONDITIONING SYSTEMS

Inside condition	Refrigerant used	System usage	Approach temperature °C	Optimal Values				Optimum ratio C_T/C_E	Penalty function value	Penalty parameter (r_k)
				ΔT_h °C	ΔT_1 °C	ΔT_{sup} °C	ΔT_{sub} °C			
	R-12	Without D.C.	4	4.6851	2.0012	1.0154	2.4445	7456.88	7456.38	0.7571331 E-
	R-12	Without D.C.	6	4.2686	2.1681	1.3740	2.0965	7837.30	7836.96	0.7748814 E-
24°C DB	R-12	Without D.C.	10	5.3756	2.1658	1.3011	1.8798	8149.68	8149.09	0.8114248 E-
65% RH	R-12	Without D.C.	14	5.8004	2.1595	1.7935	2.4925	8318.55	8318.12	0.8464056 E-3
	R-22	Without D.C.	10	4.6015	2.5301	2.0103	2.6841	8556.29	8555.89	0.8440134 E-3
	R-12	Without D.C.	6	4.4252	2.0345	1.0098	1.6621	7571.74	7571.30	0.71192785 E-3
	R-12	Without D.C.	14	4.4801	2.0418	1.0092	1.7339	8000.28	7999.93	0.79684238 E-3
27.5°C DB	R-12	Without D.C.	10	5.5781	2.1485	1.0021	2.4295	7801.22	7801.12	0.74332218 E-3
56% RH	R-12	With D.C.	10	4.5720	1.8991	2.0154	1.9901	7510.78	7510.70	0.73155271 E-3
	R-22	Without D.C.	10	4.2888	2.0462	1.0154	2.4433	8234.25	8234.13	0.92393118 E-3
	R-22	With D.C.	10	4.1410	2.0175	1.1411	1.7911	7768.50	7768.25	0.84332185 E-3

CONTINUED.....

TABLE : 4.2 : (CONTINUED)

Inside condition	Refrigerant used	System usage	Approach temperature °C	Optimal Values				Optimum ratio C_T/C_E	Penalty function value	Penalty paramete[illegible] (r_k)
				ΔT_h °C	ΔT_1 °C	ΔT_{sup} °C	ΔT_{sub} °C			
	R-12	Without D.C.	6	5.0153	2.4381	2.1101	2.3499	6671.19	6670.86	0.7956137 E-
	R-12	Without D.C.	10	5.8265	2.5198	2.5217	2.1589	6930.44	6930.31	0.6225771 E-
30°C DB 60% RH	R-12	With D.C.	10	5.1702	2.1132	2.0113	1.7245	6565.21	6565.15	0.6013766 E-
	R-12	Without D.C.	14	4.0503	2.0159	1.1352	1.6303	7347.08	7346.80	0.8332145 E-
	R-22	With D.C.	10	4.3707	2.5105	1.5361	1.8311	7150.71	7150.48	0.7954137 E-
	R-22	Without D.C.	10	4.0336	2.0077	1.0607	1.6987	7550.78	7550.56	0.7648427 E-

D.C. refers to desert cooler.

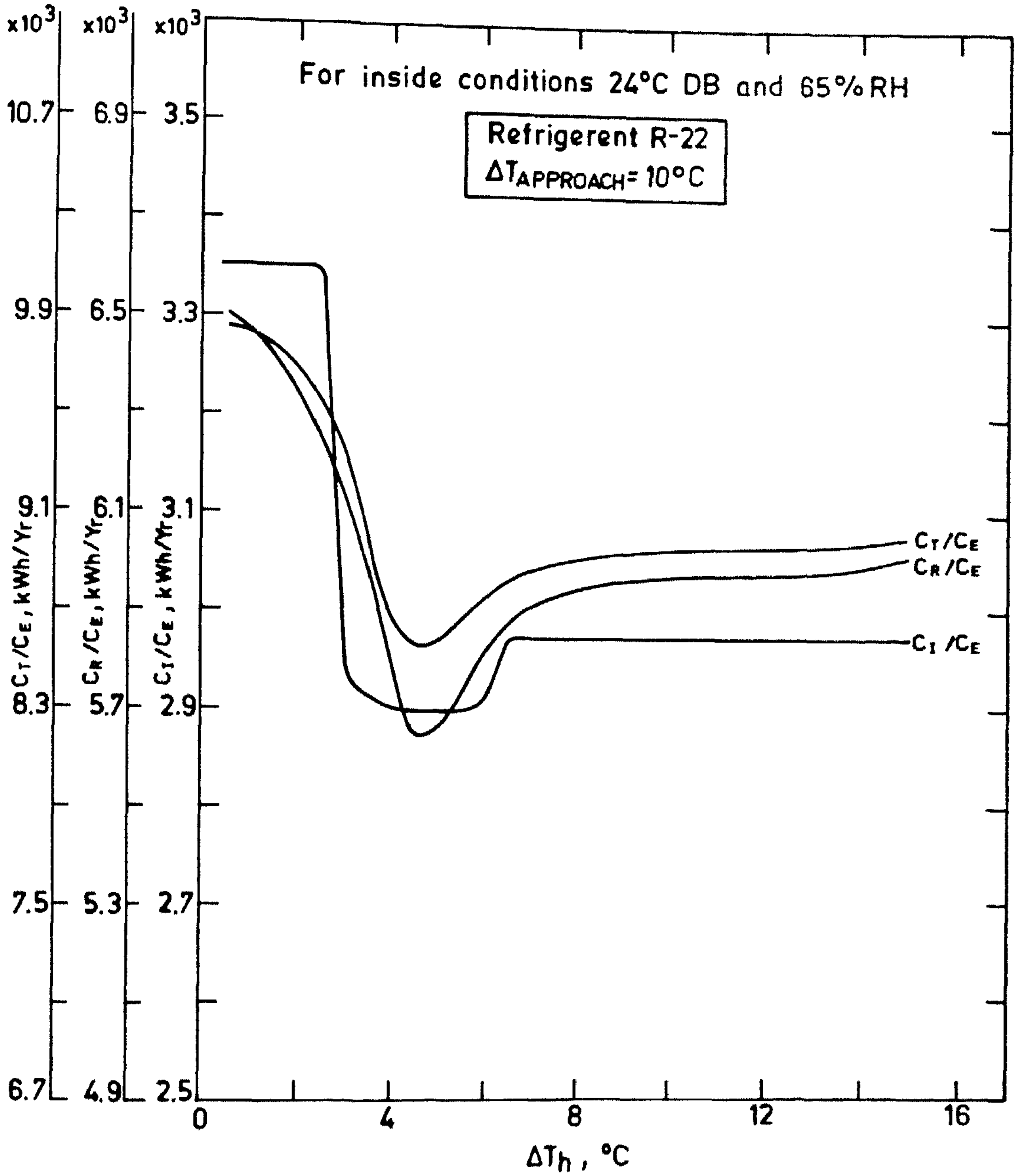

Fig. 4.4 Variation in cost with rise in condensing temperature.

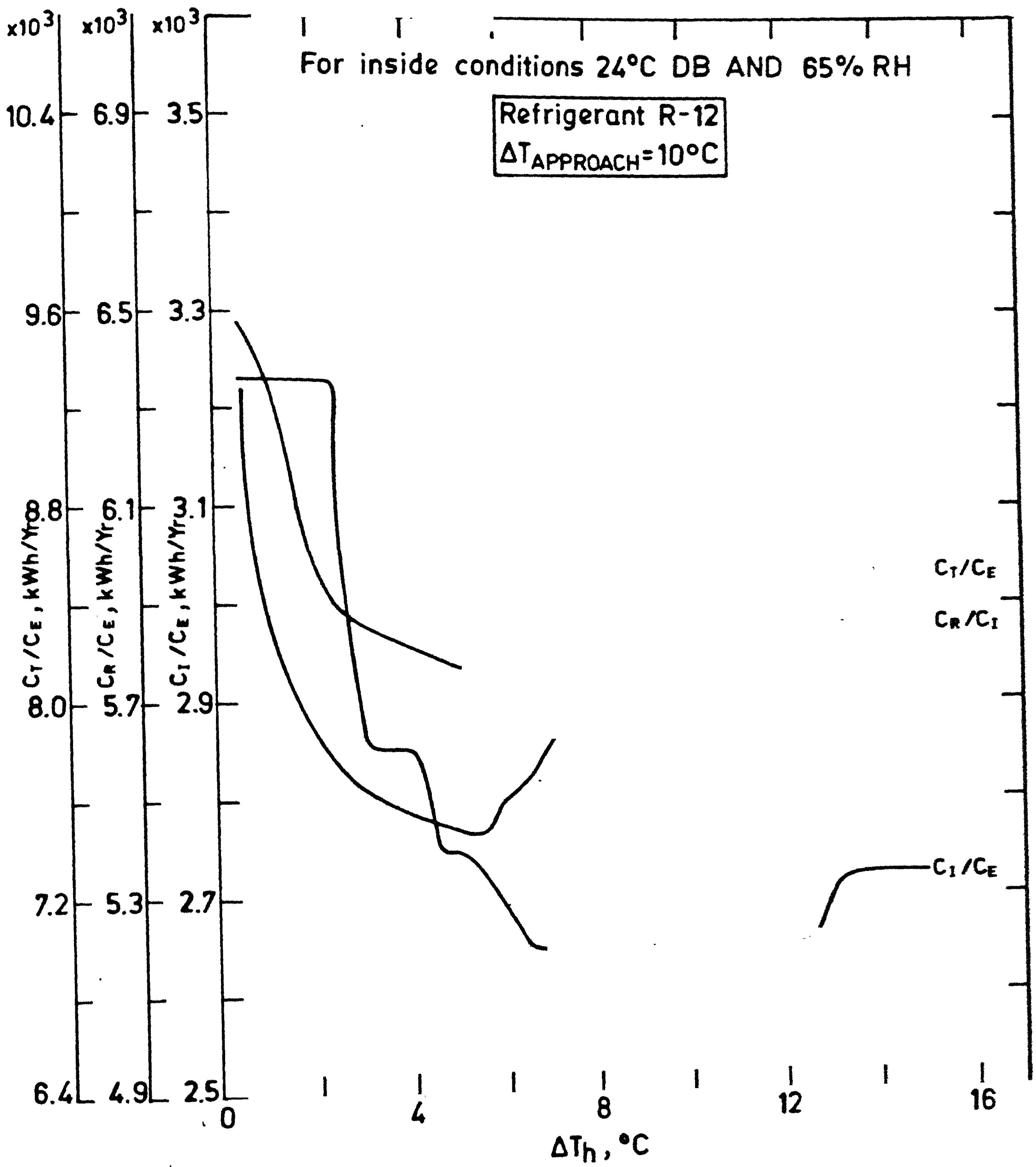

Fig. 4.5 Variation in cost with rise in condensing temperature.

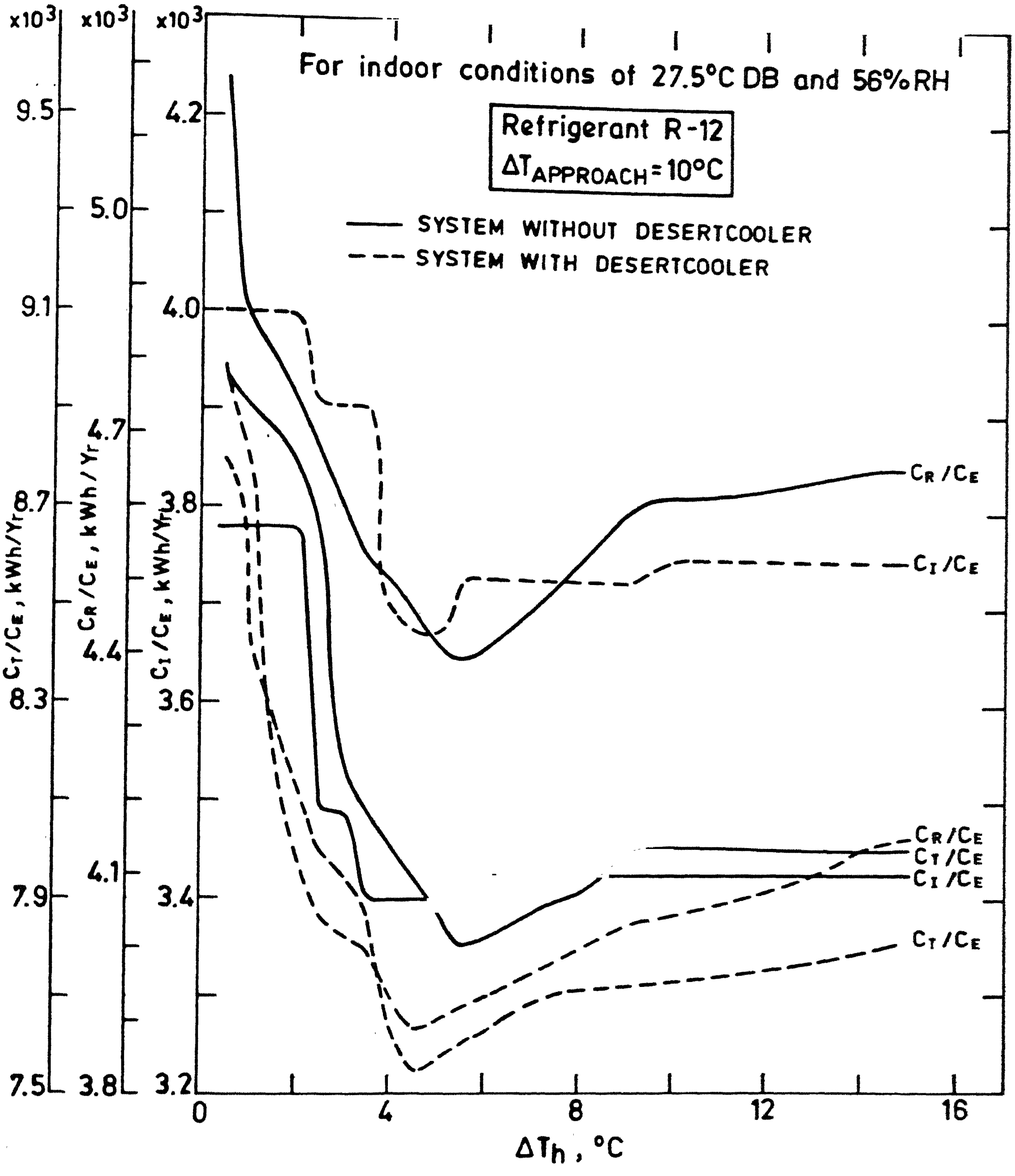

Fig. 4.6 Variation in cost with rise in condensing temperature.

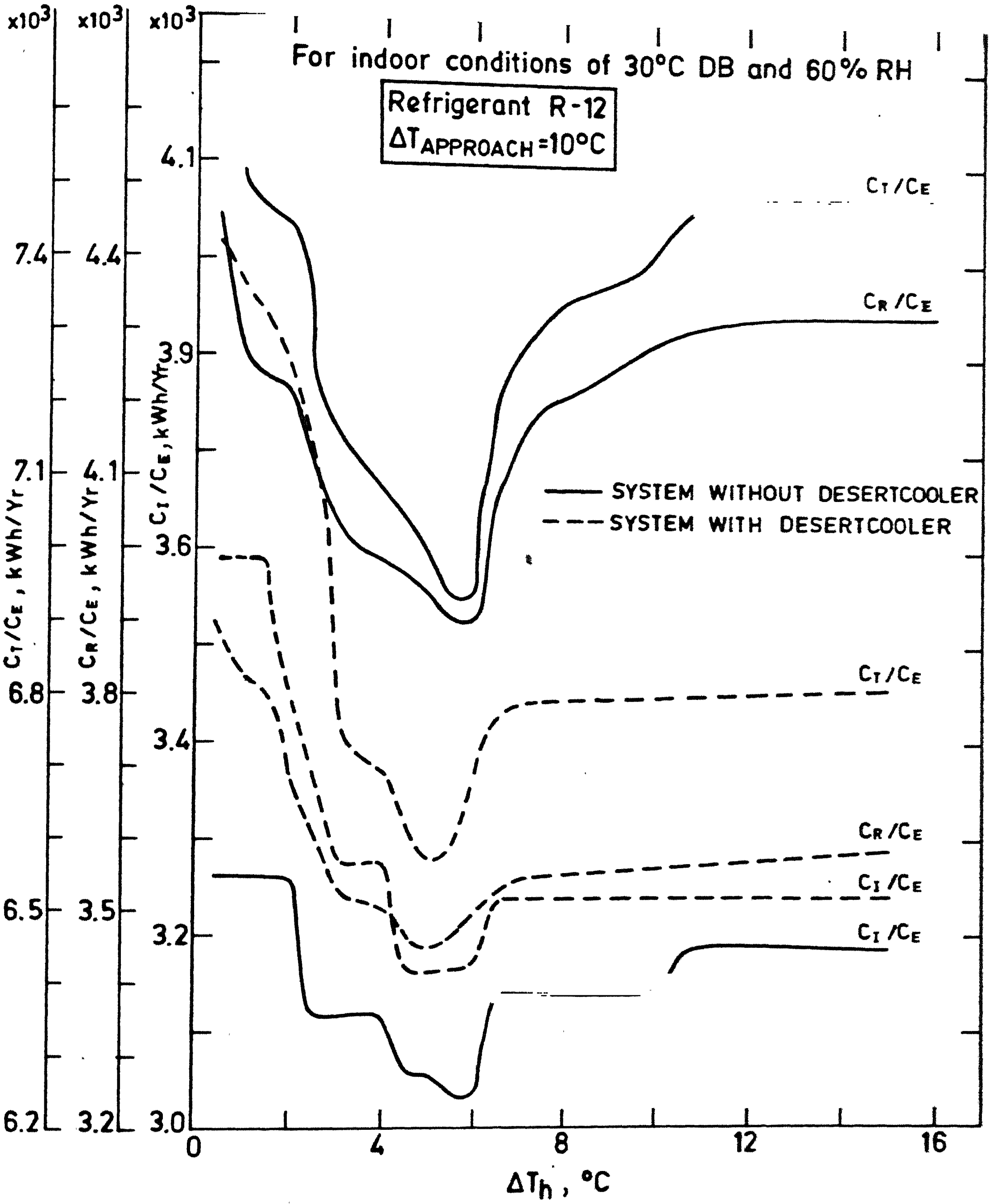

Fig. 4.7 Variation in cost with rise in condensing temperature.

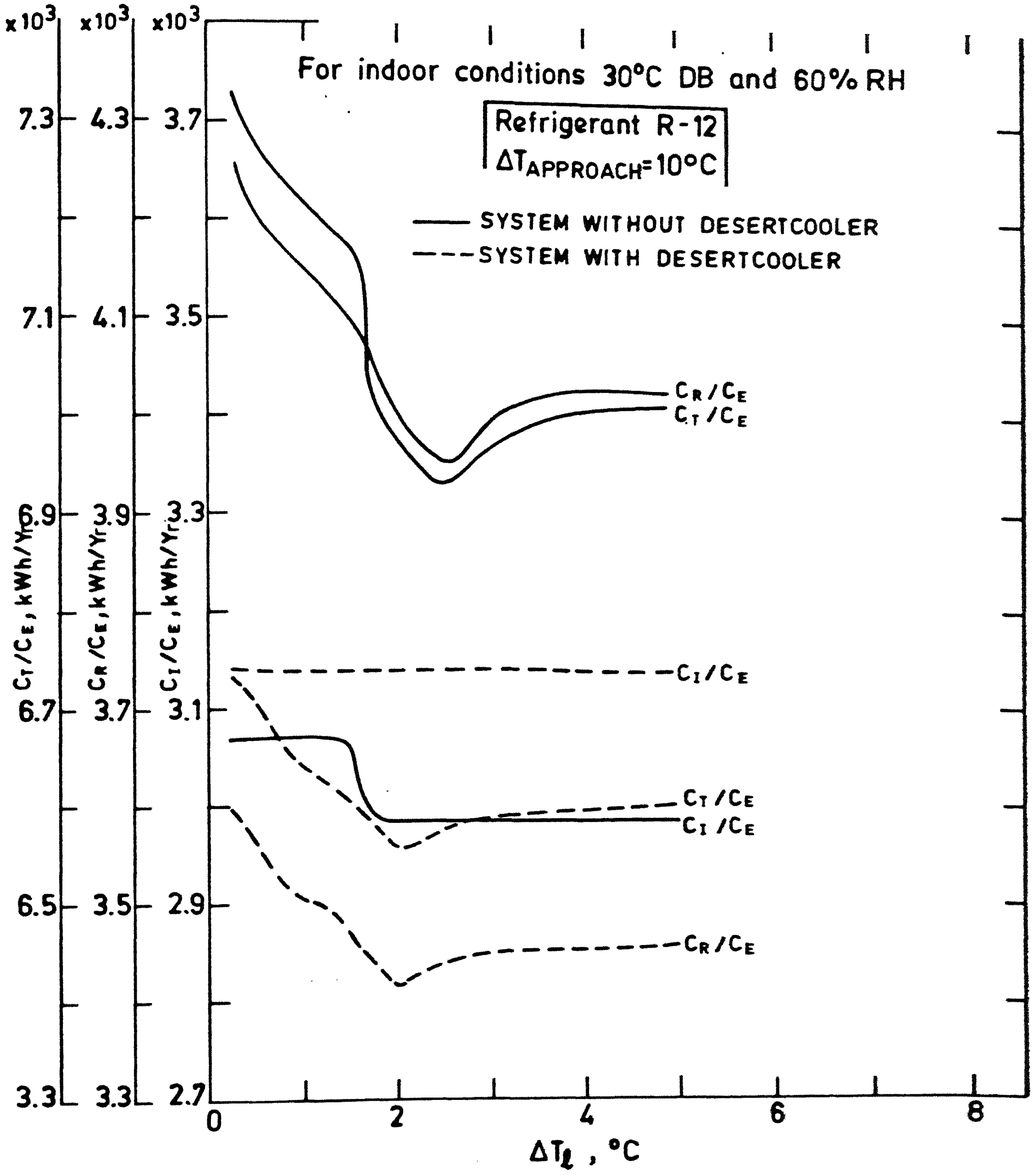

Fig. 4.8 Variation in cost with fall in evaporating temperature.

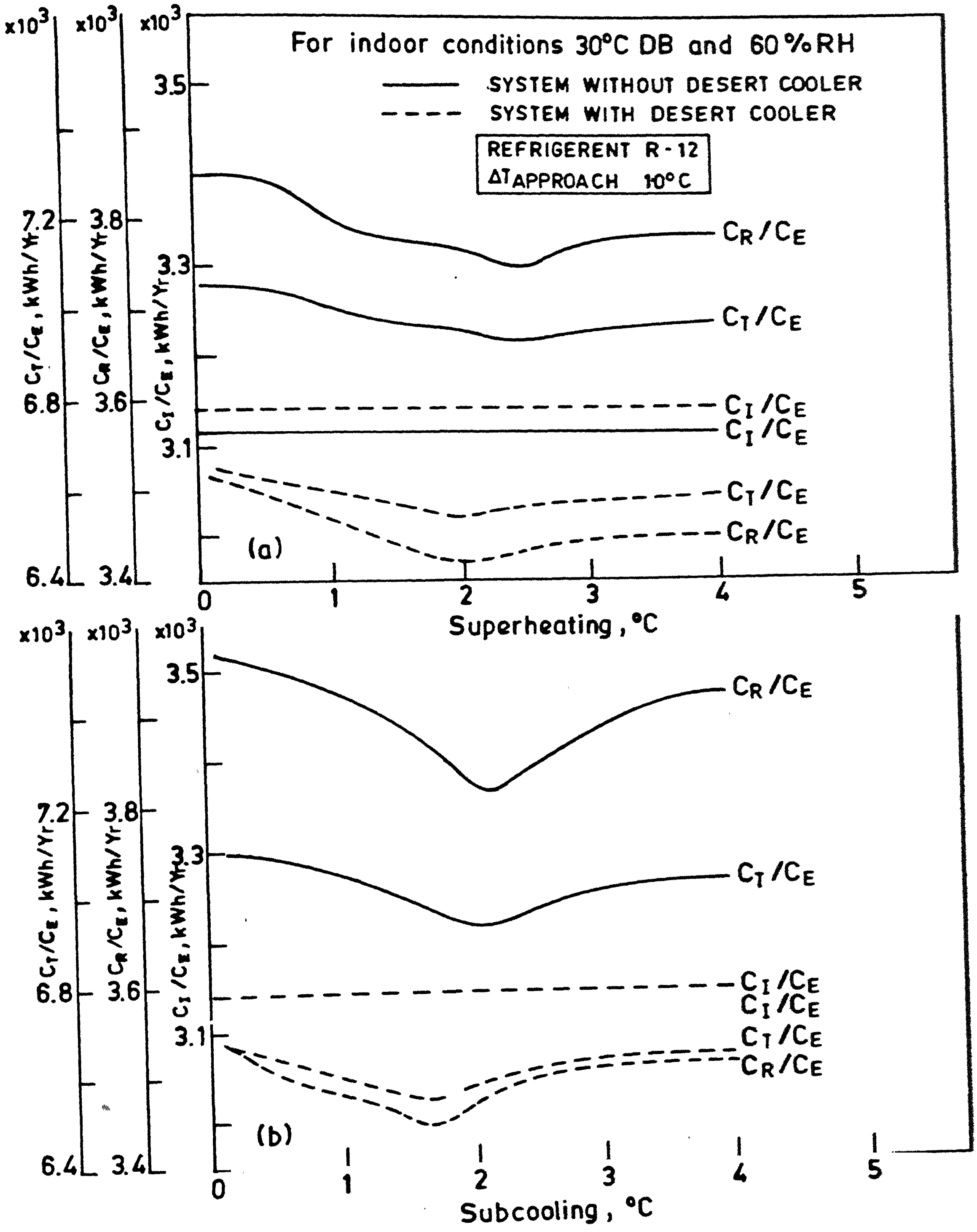

Fig. 4.9 (a) Variation in cost with degree of superheating
(b) Variation in cost with degree of subcooling.

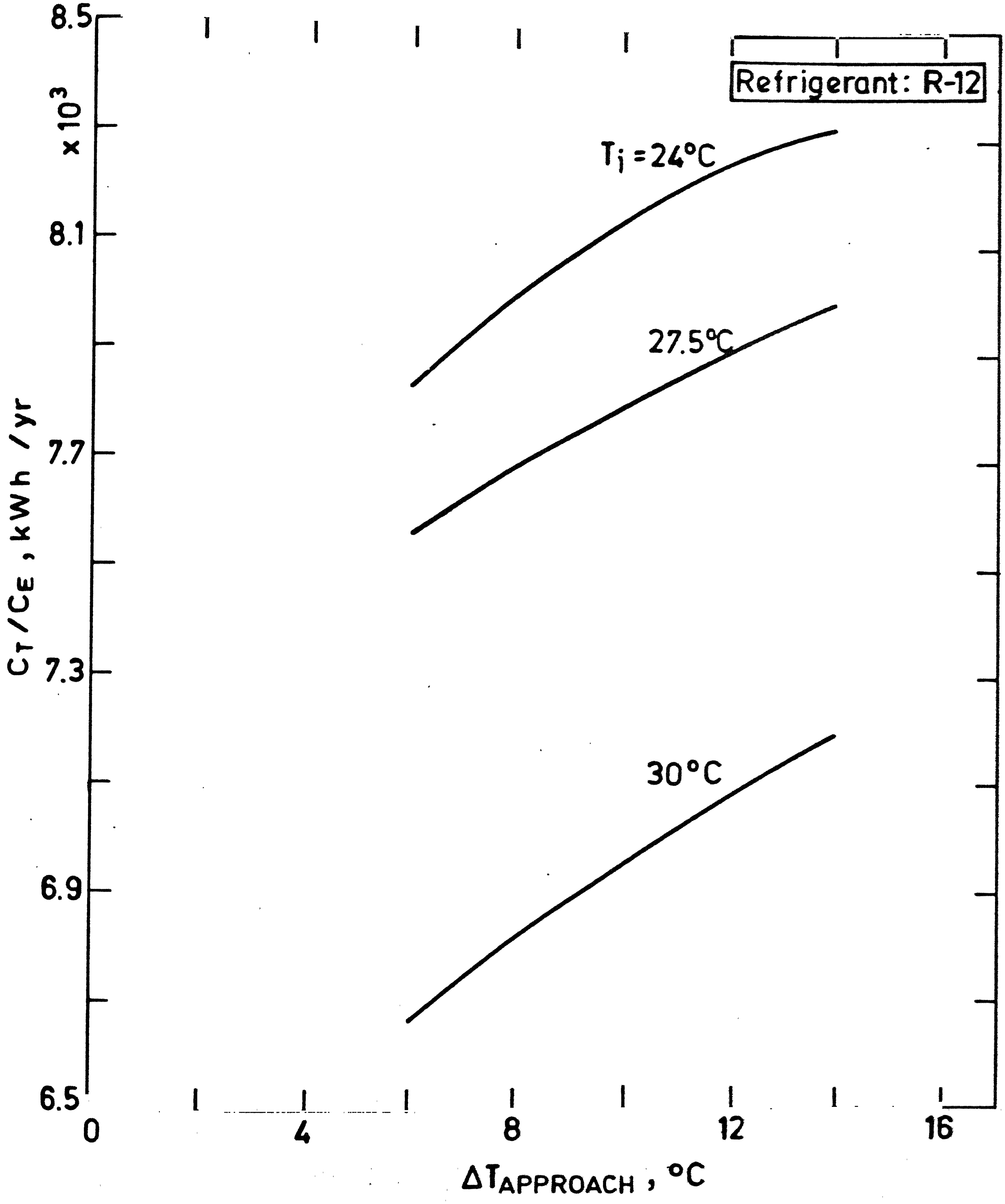

Fig. 4.10 Variation in optimum total cost with approach temperature.

CHAPTER 5

CONCLUSIONS AND SUGGESTIONS

5.1 CONCLUSIONS:

i) A multiplication factor has been found out for the estimation of actual cooling load with the help of maximum and minimum temperatures of a given location.

ii) The heat capacity and timelag factor render the cooling load about 20% less than that of without these factors.

iii) The evaporative cooling system can be substituted for the conventional air-conditioner during the period of April to June in the case of inside design condition of T_{db} = 30°C, ϕ = 60% and T_{db} = 275°C, ϕ = 56%.

iv) The implementation of the hybrid evaporative cooling and airconditioning system is found to be more economical than the conventional airconditioner, under optimum conditions.

v) For the inside conditions of 27.5°C DB, 56% RH and 30°C DB, 60% RH, the suggested hybrid system yields a gain of 4% and 5.6% over the conventional airconditioning system, respectively.

vi) The optimum total cost of the comfort airconditioning using R-12 is cheaper than that of R-22 system by 4.75% for the inside design conditions of 24°C DB and 65% RH.

vii) The variations in condensing and evaporator temperatures have more influence on the total cost of the system than the degrees of sub-cooling and superheating.

viii) The variation in total cost of the airconditioning system is almost proportional to the approach temperature.

5.2 SUGGESTIONS:

i) A combination of evaporative cooling and conventional airconditioner can be fabricated as a single unit. The overall cost can be estimated on the basis of actual cost for commercial venture.

ii) The cost of the building may also be included in the optimization in order to make the approach even more general.

iii) This approach can also be extended to the vapour-absorption system under optimum conditions.

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APPENDIX A

TABLE A-1

TEMPERATURE AND RELATIVE HUMIDITY OF THE OUTSIDE AIR FOR THE YEAR 1982, AT KANPUR

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
1	30	59	33	54	35	41	35	44	32	95	30	90	26	80
2	30	60	33	50	35	44	35	44	32	95	29	95	26	86
3	30	64	34	50	34	44	35	46	32	98	29	100	25	84
4	29	66	35	50	34	42	35	47	32	95	29	95	25	86
5	29	70	34	50	34	42	35	47	31	97	28	90	25	90
6	26	74	35	46	34	44	35	48	31	100	28	90	25	92
7	26	78	38	40	35	40	37	48	32	90	30	88	24	80
8	28	79	40	36	36	40	37	45	33	89	31	76	24	62
9	30	74	40	40	38	50	38	42	34	86	35	68	29	60
10	35	48	41	35	38	46	39	39	35	80	35	60	30	58
11	35	46	42	34	40	38	40	36	36	70	35	64	30	53
12	37	46	43	32	41	36	40	31	37	68	36	58	31	50
13	38	41	44	31	41	33	41	28	38	68	37	58	32	50
14	38	40	44	30	42	29	41	26	38	62	37	56	32	48

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APPENDIX A (TABLE A-1) (CONTINUED):

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
15	40	40	44	29	42	23	42	23	36	64	37	53	34	40
16	40	36	45	30	43	16	42	23	38	70	35	60	33	46
17	36	34	45	29	41	18	39	22	38	68	34	66	34	52
18	34	32	44	29	40	20	38	24	36	64	34	72	34	58
19	33	40	41	30	38	25	37	26	35	67	33	76	34	64
20	32	48	38	39	38	29	35	27	34	80	32	85	32	68
21	32	51	38	44	37	31	34	33	33	82	31	80	30	76
22	31	54	36	49	36	34	35	33	33	87	31	92	29	78
23	30	55	36	54	35	37	35	34	33	89	30	90	29	76
24	30	56	35	55	35	39	35	37	32	90	30	90	28	80

APPENDIX A

TABLE A-2

TEMPERATURE AND RELATIVE HUMIDITY OF THE OUTSIDE AIR FOR THE YEAR 1983, AT KANPUR

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
1	29	64	33	36	36	23	31	56	32	95	34	89	20	78
2	28	68	32	40	35	24	30	62	32	95	34	92	19	84
3	28	72	31	38	34	25	30	66	31	97	34	86	19	86
4	27	76	31	39	34	26	30	67	31	98	33	90	19	84
5	27	80	31	40	33	26	29	71	31	92	33	86	19	84
6	26	82	31	42	32	27	29	72	31	82	35	72	19	86
7	27	78	30	41	32	26	29	72	35	62	36	66	18	85
8	30	66	32	28	34	22	32	78	35	90	35	66	19	78
9	32	58	35	28	37	22	34	64	32	90	36	56	22	66
10	32	54	36	26	40	19	35	62	30	92	37	52	24	62
11	33	48	37	22	41	18	36	57	30	87	37	50	25	64
12	35	45	38	30	41	17	37	52	30	80	37	52	25	48
13	37	42	40	30	42	16	37	46	30	74	37	52	28	38
14	38	39	42	30	40	16	38	40	31	70	37	53	28	33

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APPENDIX A (TABLE A-2) : CONTINUED:

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
15	38	37	42	29	40	16	38	39	33	72	36	52	30	32
16	39	35	42	28	40	16	38	38	34	70	36	59	32	36
17	38	34	42	29	39	16	36	38	34	76	35	68	32	42
18	36	40	41	30	39	18	36	40	35	90	35	84	27	50
19	35	46	40	33	39	18	35	41	35	88	35	86	26	56
20	33	50	38	34	38	20	34	40	35	91	34	82	25	68
21	32	52	37	34	38	20	33	49	34	92	34	82	25	74
22	31	56	36	36	37	22	32	52	33	95	34	84	22	76
23	31	58	35	36	37	23	32	56	32	92	34	87	22	74
24	30	60	34	36	36	23	31	56	32	92	34	86	22	74

APPENDIX A

TABLE A-3

TEMPERATURE AND RELATIVE HUMIDITY OF THE OUTSIDE AIR FOR THE YEAR 1985, AT KANPUR

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
1	28	37	32	44	33	42	32	69	30	76	32	68	25	76
2	28	38	30	44	32	39	32	76	30	78	31	72	25	80
3	27	39	29	47	32	38	32	74	30	80	30	71	24	83
4	26	40	29	51	32	37	32	78	30	83	30	72	24	82
5	26	40	28	54	29	36	31	78	31	82	29	76	24	89
6	28	44	28	56	29	39	31	66	31	84	29	78	23	86
7	30	48	29	56	31	36	32	70	33	72	30	75	24	89
8	31	42	32	54	33	33	34	81	34	64	34	68	26	74
9	32	30	36	44	35	27	35	64	36	60	37	56	28	44
10	34	30	41	38	37	24	36	84	37	59	37	52	29	30
11	36	26	43	32	39	22	37	70	38	52	38	48	30	22
12	38	23	43	26	39	20	39	52	39	50	38	45	31	23
13	40	22	44	24	40	18	39	57	38	48	38	43	31	23
14	38	20	45	22	41	15	40	50	36	48	37	43	31	23

CONTINUED.....

APPENDIX A (TABLE A-3) (CONTINUED):

Time in hrs	April		May		June		July		August		September		October	
	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %	T °C	Ø %
15	37	20	44	20	40	15	40	46	35	46	38	42	30	20
16	34	20	45	20	40	15	35	48	34	45	37	41	29	25
17	32	20	44	23	39	16	34	52	34	44	35	42	29	29
18	29	22	39	26	39	20	34	61	32	52	34	52	29	42
19	29	24	37	29	38	26	33	66	33	58	33	58	28	50
21	29	25	36	32	38	28	33	70	31	62	33	60	26	60
22	28	27	35	35	37	32	33	72	30	78	32	64	24	64
23	27	33	32	40	34	36	32	72	31	75	32	68	23	70
24	26	35	32	42	32	38	32	70	30	75	32	68	22	72

APPENDIX - B

APPENDIX-B

Table-B

VALUES FOR SOLAR INTENSITY CALCULATIONS

Month.	Equation of time/minutes	Declination of angle degrees	A_1 W/m^2	B_1	C_1
Jan.	-11.2	-20.0	1230	0.142	0.058
Feb.	-13.9	-10.8	1214	0.144	0.060
March	- 7.5	0.0	1185	0.156	0.071
April	1.1	11.6	1135	0.180	0.097
May	3.3	20.0	1103	0.196	0.121
June	-1.4	23.45	1088	0.205	0.134
July	-6.2	20.6	1085	0.207	0.136
August	-2.4	12.3	1107	0.201	0.122
September	7.5	0.0	1151	0.177	0.092
October	15.4	-10.6	1192	0.160	0.073
November	13.8	-19.8	1220	0.149	0.063
December	1.6	-23.45	1233	0.142	0.057

A_1 = Apparant solar irradiation at air mass = 0 , W/m^2

B_1 = Atmospheric extinction coefficient

C_1 = Diffuse radiation factor.

APPENDIX - C

APPENDIX - C

EQUATIONS FOR THE ESTIMATION OF PRIMARY ANGLES

The equation for, LST, local solar time

$$LST = LCT + \text{Equation of time} \quad (C-1)$$

where,

$$LCT = IST \pm (82.5 - \text{Longitude of the location})/15, \text{ hrs} \quad (c-2)$$

The equations for the hour angle α_h, are given by

$$\alpha_h = (LST - 12.0).15, \text{ for } LST < 12.0$$
$$= (12.0 - LST).15, \text{ for } LST > 12.0 \quad (C-3)$$

The relationships between the latitude angle l, the hour angle α_h, declination angle d, altitude angle β, and azimuth angle γ are,

$$\sin\beta = (\cos l.\ \cos h.\ \cos d + \sin l.\ \sin d) \quad (C-4)$$

$$\cos\gamma = \sin\ (\cos l.\ \sin d - \cos d.\ \sin l.\ \cos\alpha_h) \quad (C-5)$$

For wall solar azimuth angle α,

$$\alpha = \left|\frac{\pi}{2} - \gamma\right|, \text{ for east and west facing walls,}$$
$$= \gamma \quad , \text{ for north facing walls,}$$
$$= |(\gamma - \pi)|, \text{ for south facing walls,} \quad (C-6)$$

The relationship for the incidence angle, Θ, is

$$\text{Cos}\Theta = (\text{Cos}\beta)(\text{Cos}\alpha)(\text{Cos}\phi_t) + (\text{Sin}\ )(\text{Sin}\ \phi) \text{ for}$$

tilted surfaces, with tilt angle ϕ_t

$$= (\text{Cos}\beta)(\text{Cos}\alpha), \text{ for vertical surfaces}$$

$$= \text{Sin}\,\beta, \text{ for horizontal surfaces.} \qquad \text{(C-7)}$$

APPENDIX - D

APPENDIX-D

EQUATIONS FOR THERMAL PROPERTIES OF GLASS

The relationships for the thermal properties of the glass, transmittivity, t_r, reflectivity r_g and absorptivity, a_g,

$$t_r = (1-r')^2 \cdot a_c/(1 - r'^2 \cdot a_c^2) \qquad \text{(D-1)}$$

$$r_g = r' + \frac{(1 - r')^2 \cdot a_c^2}{(1 - r'^2 \cdot a_c^2)} \qquad \text{(D-2)}$$

$$a_g = 1 - r' - \frac{(1 - r')^2 \cdot a_c}{(1 - r' \cdot a_c)} \qquad \text{(D-3)}$$

where,

$$r' = \frac{1}{2}\left[\frac{\mathrm{Sin}^2(\theta_g - \theta_g')}{\mathrm{Sin}^2(\theta_g + \theta_g')} + \frac{\tan^2(\theta_g - \theta_g')}{\tan^2(\theta_g - \theta_g')}\right]$$

$$a_c = \exp^{-E_c x_g/(1 - \frac{\mathrm{Sin}^2 \theta_g}{n^2})}$$

where θ_g Angle of incidence

θ_g' Angle of refracted rays

E_c Extinction coefficient

x_g Thickness of the glass (m)

n Refraction index.

APPENDIX-E

APPENDIX-E

BUILDING PARTICULARS FOR COOLING LOAD CALCULATIONS

Room Size = 8.4 x 4.0 x 3.8, m^3

Thickness of the wall = 0.4 m

Thickness of the roof = 0.2 m

Number of windows:

- On the Northern wall = 2
- On the Southern wall = 2
- On the Eastern wall = 0
- On the Western wall = 0

Area of the windows = 1.08 m^2

Number of doors:

- On Western walls = 1
- On the other sides = 0

Number of occupants = 25

Number of fans inside the room = 4

Number of lights inside the room = 8

Ratings:

- Fans = 100 watts
- Lights = 40 watts.

APPENDIX-F

APPENDIX-F

PROPERTIES OF REFRIGERANTS

Refrigerant R-12:

$$c_p(T) = 0.59524 + 0.00181715\,(T+15)$$

$$s_f(T) = 0.142093 + 0.33747\,(T/100) - 0.0399606(T/100)^2 + 0.019868(T/100)^3 + 0.014023\,(T/100)^4$$

$$h_g(T) = 188.86 + 0.440278(T) - 7.02007(T/100)^2 - 5.07651\,(T/100)^3 - 3.82545\,(T/100)^4$$

$$h_f(T) = 36.1554 + 0.928108(T) + 6.89916\,(T/100)^2 + 3.73414\,(T/100)^3 + 5.91673\,(T/100)^4$$

Refrigrant R-22:

$$c_p(T) = 0.60 + 0.0005246\,(T+45+0.043686\,(T+45)^2)),$$

for $-60 < T < 10$

$$= 0.70114 + 0.0029529\,(T-10+0.010472(T-10)^2)$$

for $10 < T < 50$

$$s_f(T) = 0.181332 + 0.43667\,(T/100) - 0.0596756\,(T/100)^2 - 0.0234343\,(T/100)^3 + 0.0638047\,(T/100)^4$$

$$h_g(T) = 251.06 + 35.4577\,(T/100) - 19.4993\,(T/100)^2 - 7.62005\,(T/100)^3 - 11.6756\,(T/100)^4$$

$$h_f(T) = 46.2102 + 1.20394\,(T) + 6.828\,(T/100)^2 - 8.99387\,(T/100)^3 + 20.0612\,(T/100)^4$$

where, T in °C, c_p in kJ/kg.°C, s_f in kJ/kg.°K, h_g, h_f in kJ/kg.

APPENDIX - G

```
     1 COOLINGLOADS FOR THE MONTH OF JULY',3X,'|',9X,'|',/,31X,
     172('*'))
      GO TO 33
27    WRITE(3,28)
28    FORMAT(31X,72('*'),/,31X,'|',5X,'|',7X,'|',1X,'VARIOUS
     1 COOLINGLOADS FOR THE MONTH OF AUGUST',1X,'|',9X,'|',/,31X,
     172('*'))
      GO TO 33
29    WRITE(3,30)
30    FORMAT(31X,72('*'),/,31X,'|',5X,'|',7X,'|',1X,'VARIOUS
     1 COOLINGLOADS FOR THE MONTH OF SEPTEMBER',9X,'|',/,31X,72('*'))
      GO TO 33
31    WRITE(3,32)
32    FORMAT(31X,72('*'),/,31X,'|',5X,'|',7X,'|',1X,'VARIOUS
     1 COOLINGLOADS FOR THE MONTH OF OCTOBER','|',9X,'|',/,31X,
     172('*'))
33    WRITE(3,34)
34    FORMAT(31X,'|',1X,'IST',1X,'|',2X,'LST',2X,'|','Qsolar','|',
     12X,'Qglass',1X,'|',2X,'Qinfil',1X,'|',2X,'Qvent',2X,'|',2X,
     1'Oequip',1X,'|',2X,'Qtotal',1X,'|',/31X,72('*'))
*-------------------Calculation of the solar intensity----------------------*
      IST=0.0
      TOSOE=0.0;TOSOW=0.0;TOSON=0.0;TOSOS=0.0;TOSOH=0.0
      K=0.0
35    K=K+1
      IST=IST+1
      AAIN=0.0
      CAL=0.0
      GGAMA=0.0
      RRISE=0.0
      SETS=0.0
      BETAA=0.0
      DO 45 J=1,N
      DEC=D(I,J)*C
      H=-(SIN(ALT)*SIN(DEC))/(COS(ALT)*COS(DEC))
      DEG=ACOS(H)/C
      SET=(180.0+DEG)/15.0
      RISE=12.0-DEG/15.0
      IF(82.5-ALONG)37,36,36
36    ALCT=IST-(82.5-ALONG)*4.0/60.0
      GO TO 38
37    ALCT=IST+(82.5-ALONG)*4.0/60.0
38    ALST=ALCT+EQT(I,J)/60.0
      DIFF=ALST-12.0
      IF(DIFF)39,40,40
39    AH=((12.0-ALST)*60.0)*C/4.0
      GO TO 41
40    AH=((ALST-12.0)*60.0)*C/4.0
41    B=COS(ALT)*(ABS(COS(AH)))*COS(DEC)+SIN(ALT)*SIN(DEC)
      AIN=(X(I,J)/EXP(Y(I,J)/B))/1000.0
      BET=ASIN(B)
      G=(COS(ALT)*SIN(DEC)-COS(DEC)*SIN(ALT)*ABS(COS(AH)))/COS(BET)
      BETA=BET/C
      GAM=ACOS(G)/C
      IF(DIFF)43,42,42
42    ANEGAM=(180.0-GAM)
      GAMA=ANEGAM
      GO TO 44
43    GAMA=GAM
44    AINT=AIN
      GAM=GAMA
      SSET=SET
      RISES=RISE
      BET=BETA
```

```
      AAIN=AINT+AAIN
      CAL=ALST+CAL
      GGAMA=GAM+GGAMA
      RRISE=RISES+RRISE
      SETS=SSET+SETS
      BETAA=BET+BETAA
15    CONTINUE
      AIN=AAIN/4.0
      GAMA=GGAMA/4.0
      RISE=RRISE/4.0
      SET=SETS/4.0
      ALST=CAL/4.0
      BETA=(BETAA/4.0)*C
      P=Z(I)
      TMAX=TM(I)
      TMIN=TMN(I)
      PHI=ALOG(P)
      ANLST=ALST-PHI
      IF(ANLST.LE.0.0)ANLST=24+ANLST
      AREAW=ALENG*HEIG-HW*BW*NW
      AREAE=ALENG*HEIG-HW*BW*NE
      AREAN=BRE*HEIG-HW*BW*NN
      AREAS=BRE*HEIG-HW*BW*NS
      AREAH=ALENG*BRE
      CALL TODISN(TMAX,TMIN,RISE,IST,C,TO)
      CALL SOLAIR(AIN,BETA,GAMA,RISE,SET,ALST,P,C,VEL,TO,ABST,SOLAE,
     1SOLAW,SOLAN,SOLAS,SOLAH,HO,HOH,ALPS,ALPN,DN,DS,DIFE,DAINN,
     1DAINS)
      WRITE(2,45)IST,ALST,TO,SOLAE,SOLAW,SOLAN,SOLAS,SOLAH
45    FORMAT(31X,'|',1X,I2,2X,'|',1X,F5.2,1X,'|',1X,F5.2,'|',
     15(2X,F5.2,2X,'|'))
      TOSOE=SOLAE-TI+TOSOE
      TOSOW=SOLAW-TI+TOSOW
      TOSON=SOLAN-TI+TOSON
      TOSOS=SOLAS-TI+TOSOS
      TOSOH=SOLAH-TI+TOSOH
      CALL SOLAIR(AIN,BETA,GAMA,RISE,SET,ANLST,P,C,VEL,TO,ABST,SOLNE,
     1SOLNW,SOLNN,SOLNS,SOLNH,HN,HNR,ALS,ALN,XX,XIX,DIXE,DAXN,DAXS)
      SOLANE(K)=SOLNE;SOLANW(K)=SOLNW;SOLANN(K)=SOLNN;SOLANS(K)=SOLNS
      SOLANH(K)=SOLNH;TOUT(K)=TO;ST(K)=ALST
      IF(IST.LE.23.0)GO TO 48
      EXME=TOSOE/24.0
      EXMW=TOSOW/24.0
      EXMS=TOSOS/24.0
      EXMN=TOSON/24.0
      EXMH=TOSOH/24.0
      DFACT=0.1
      K=0
47    K=K+1
      VROOM=8.4*4.0*3.8
*-|-|-|-|--Calculation of infiltration & ventilation loads-------------*
      CALL MISC(VROOM,TI,TOUT(K),QIN,QVEN,SENINF,SENVEN)
      CALL COHTCT(TI,TOUT(K),VEL,U)
      CALL COHTRF(TI,TOUT(K),VEL,UR)
*-|-|-|-|------Calculation of the heat transfer through structure---------(*
      QSOLAR(K)=U*(EXME+DFACT*(SOLANE(K)-EXME-TI))*AREAE
      QSOLAR(K)=QSOLAR(K)+U*AREAW*(EXMW+DFACT*(SOLANW(K)-EXMW-TI))
      QSOLAR(K)=QSOLAR(K)+U*AREAS*(EXMS+DFACT*(SOLANS(K)-EXMS-TI))
      QSOLAR(K)=QSOLAR(K)+U*AREAN*(EXMN+DFACT*(SOLANN(K)-EXMN-TI))
      QSOLAR(K)=QSOLAR(K)+UR*AREAH*(EXMH+DFACT*(SOLANH(K)-EXMH-TI))
      QINFIL(K)=QIN;VENTIL(K)=QVEN;SENQIN(K)=SENINF;SENVET(K)=SENVEN
      QTOGLS(K)=HW*BW*WN*(QGLSN+QGLSS)
      QEQUIP(K)=NF*RATFAN+NL*RATLIT
      IF(K.LE.23.0)GO TO 47
```

```
      GO TO 49
*---------Calculation of the heat transfer through glass windows---------*
 48   CALL PROGLS(C,DN,BETA,ALPN,FSN,THOUN,ALPHAN)
      CALL PROGLS(C,DS,BETA,ALPS,FSS,THOUS,ALPHAS)
      HOUT=(3.98*1.0E-03+9.392*1.0E-04*VEL)
      HIN=(0.237*((VIN/H)**0.5))/3600
      AIRN=RHO*DAINN
      AIRS=RHO*DAINS
      UGL=1/(1/HIN+1/HOUT)
      QGLSN=FSN*THOUN*DAINN+THOUD*DIFE+AIRN*THOUN+(FSN*ALPHAN*DAINN+
     1ALPHAD*DIFE+ALPHAN*AIRN)/(1+HOUT/HIN)+UGL*(TOUT(K)-TI)
      QGLSS=FSS*THOUS*DAINS+THOUD*DIFE+AIRS*THOUS+(FSS*ALPHAS*DAINS+
     1ALPHAD*DIFE+ALPHAS*AIRS)/(1+HOUT/HIN)+UGL*(TOUT(K)-TI)
      QOCCUP=NOCCUP*QACT/3600.0
      IF(IST.LE.23.0)GO TO 35
 49   DO 54 K=1,24
      IF(ST(K).GT.RISE.AND.ST(K).LT.SET)GO TO 50
*---------Calculation of the total cooling load--------------------------*
      QSOLAR(K)=0.0;QTOGLS(K)=0.0
 50   QTOTAL(K)=QSOLAR(K)+QTOGLS(K)+QINFIL(K)+VENITL(K)+QEQUIP(K)
      QTOTAL(K)=1.1*(QTOTAL(K)+QOCCUP)
      IF(K.GE.8.AND.K.LE.17) GO TO 51
      GO TO 52
 51   QRUN=QTOTAL(K)+QRUN
 52   RSH=1.1*(QSOLAR(K)+QTOGLS(K)+SENQIN(K)+SENVET(K)+QOCCUP)
      ERSH=RSH/QTOTAL(K)
      WRITE(3,53)K,ST(K),QSOLAR(K),QTOGLS(K),QINFIL(K),VENTIL(K)
     1,QEQUIP(K),QTOTAL(K)
 53   FORMAT(31X,'I',1X,I2,2X,'I',1X,F5.2,1X,'I',1X,F5.2,'I',5(2X,F5.2
     1,2X,'I'))
 54   CONTINUE
      RH1=RH*100
      TYPE*,RH1
      WRITE(2,55)RISE,SET,TI,RH1
 55   FORMAT(31X,72('*'),/,31X,'I',9X,'SUNRISE=',F5.2,'Hrs',20X,
     1'SUNSET=',F5.2,'Hrs',10X,'I',/,31X,'I',2X,'ROOM TEMPERATURE=',
     2F4.1,1X,'Deg Celcius',4X,'RELATIVE HUMIDITY=',F4.1,'%',8X,
     3'I',/,31X,72('*'),/)
      WRITE(3,56)RISE,SET,TI,RH1
 56   FORMAT(31X,72('*'),/,31X,'I',9X,'SUNRISE=',F5.2,'Hrs',20X,
     1'SUNSET=',F5.2,'Hrs',10X,'I',/,31X,'I',2X,'ROOM TEMPERATURE=',
     2F4.1,1X,'Deg Celcius',4X,'RELATIVE HUMIDITY=',F4.1,'%',8X,
     3'I',/,31X,72('*'))
      WRITE(3,57)QRUN
 57   FORMAT(31X,'TOTAL COOLING LOAD FOR THE RUNNING HOURS ( from
     18.00 A.M to 5.00 P.M)=',F5.2,1X,'K.W',/)
 58   CONTINUE
      STOP;END
*-----------------------------------------------------------------------*
*       SUBROUTINE FOR OUTSIDE AIR TEMPERATURE VARIATION                *
*-----------------------------------------------------------------------*
      SUBROUTINE TODISN(TMAX,TMIN,R,I,C,TO)
      B=(TMAX-TMIN)/(1-COS((15*(R-1.0)-15*(R+11.0))*C))
      A=TMAX-B
      D=15*(R+11.0)
      TO=A+B*COS((15*I-D)*C)
      RETURN;END
*-----------------------------------------------------------------------*
*              SUBROUTINE FOR SOL-AIR TEMPERATURES                      *
*-----------------------------------------------------------------------*
      SUBROUTINE SOLAIR(A,B,G,R,S,ALST,Z,C,VEL,TO,A3,SOLAE,SOLAW,SOLAN
     1,SOLAS,SOLAH,HO,HOH,ALPS,ALPN,DN,DS,DIFE,DAINN,DAINS)
      IF(ALST.GT.R.AND.ALST.LT.S)GO TO 1
      SOLAE=TO;SOLAW=TO;SOLAS=TO;SOLAN=TO;SOLAH=TO
```

```
      GO TO 2
1     ALPE=(ABS(90-G))*C
      D=COS(B)*COS(ALPE)
      DAINE=A*D
      DIFE=Z*A*0.5
      IF(ALST.GE.12.0.AND.ALST.LE.6)DAINE=0.0
      TOTINE=DAINE+DIFE
      DAINW=A*D
      IF(ALST.GT.R.AND.ALST.LE.12.0)DAINW=0.0
      TOTINW=DAINW+DIFE
      ALPN=G*C
      DN=COS(B)*COS(ALPN)
      DAINN=A*DN
      IF(ALST.GT.12.0)DAINN=0.0
      TOTINN=DAINN+DIFE
      ALPS=(ABS(G-180.0))*C
      IF(ALST.GT.12.0)ALPS=(ABS(G+180.0))*C
      DAINS=0.0
      IF(ALPS.LT.3.1415926/2.0)DAINS=A*COS(B)*COS(ALPS)
      TOTINS=DAINS+DIFE
      DIFH=Z*A
      TOTINH=A*SIN(B)+DIFH
      HO=(18.423+3.81*VEL)/3600
      HOH=(4.1897*(6.34+1.21*VEL))/3600
      SOLAE=TO+AB*TOTINE/HO
      SOLAW=TO+AB*TOTINW/HO
      SOLAN=TO+AB*TOTINN/HO
      SOLAS=TO+AB*TOTINS/HO
      SOLAH=TO+AB*TOTINH/HOH
2     DN=COS(B)*COS(ALPN)
      DS=COS(B)*COS(ALPS)
      RETURN;END
*-------------------------------------------------------------------------*
*   SUBROUTINE FOR  CONVECTIVE HEAT TRANSFER COEFFICIENT(walls)           *
*-------------------------------------------------------------------------*
      SUBROUTINE CONTCT(TI,TO,VEL,U)
      HO=(18.423+3.48*VEL)/3600
      X1=0.025;X2=0.15;X3=0.025
      AK1=0.00072;AK2=0.0013
      TCOND=X1/AK1+X2/AK2+X3/AK1
      C1=0.00118055
      C2=1/HO+TCOND
      DT=10.0
1     FN=C1*C2*(DT**1.25)+DT-TO+TI
      FDN=1.25*C1*C2*(DT**0.25)+1.0
      VAL=FN/FDN
      DTNEW=ABS(DT-VAL)
      IF(ABS(DTNEW-DT).LT.0.01)GO TO 2
      DT=DTNEW
      GO TO 1
2     HI=C1*(DT)**0.25
      U=1/(1/HI+TCOND+1/HO)
      RETURN;END
*-------------------------------------------------------------------------*
*   SUBROUTINE FOR  CONVECTIVE HEAT TRANSFER COEFFICIENT( roof)           *
*-------------------------------------------------------------------------*
      SUBROUTINE CONTRF(TI,TO,VEL,JR)
      HOH=(4.1897*(6.34+1.21*VEL))/3600
      X1=0.015;X2=0.07;X3=0.015
      AK1=0.00072;AK2=0.0013
      TCONR=X1/AK1+X2/AK2+X3/AK1
      C1=3.48/3600.0
      C2=1/HOH+TCONR
      DT=10.0
```

```
1     FN=C1*C2*(DT**1.25)+DT-TO+TL
      FDN=1.25*C1*C2*(DT**0.25)+1.0
      VALR=FN/FDN
      DTNEW=ABS(DT-VALR)
      IF(ABS(DTNEW-DT).LT.0.01)GO TO 2
      DT=DTNEW
      GO TO 1
2     HIR=C1*(DT**0.25)
      UR=1/(1/HIR+TCONR+1/HOH)
      RETURN;END
*----------------------------------------------------------------------------*
            SUBROUTINE FOR PROPERTIES OF GLASS                               *
*----------------------------------------------------------------------------*
      SUBROUTINE PROGLS(C,DX,BETA,ALP,FS,THOU,ALPHA)
      H=1.2;B=0.9;D=0.3
      AL=0.0064;AK=0.174/0.0254;AN=1.526
      R1=D/H
      R2=D/B
      AIC=ACOS(DX)
      DELTA=ABS(ATAN(SIN(BETA)/(COS(BETA)*COS(ALP))))
      IF(DELTA.GT.22/14)DELTA=DELTA-22/14
      Y=R1*(ABS(SIN(DELTA)/COS(DELTA)))
      Z=R2*(ABS(SIN(ALP)/COS(ALP)))
      IF(Y.OR.Z.GT.1.0)GO TO 1
      FS=1-Y-Z+Y*Z
      GO TO 2
1     FS=0.0
2     X=SQRT(1-SIN(AIC)**2.0/(AN*AN))
      ALN=AL/X
      ABS=EXP(-AK*ALN)
      THETA=ACOS(X)
      ADD=THETA+AIC
      ANEW=AIC-THETA
      REFY=(SIN(ANEW)**2.0/SIN(ADD)**2.0+((SIN(ANEW)**2.0)*(COS(ADD)**
     12.0))/((COS(ANEW)**2.0)*(SIN(ADD)**2.0)))/2.0
      THOU=((1-REFY)**2.0)*ABS/(1-(REFY*ABS)**2.0)
      ALPHA=1-REFY-((1-REFY)**2.0)*ABS/(1-REFY*ABS)
      RETURN;END
*----------------------------------------------------------------------------*
*           SUBROUTINE FOR THE MISCELLANEOUS LOAD                            *
*----------------------------------------------------------------------------*
      SUBROUTINE MISC(VROOM,TI,TO,QI,QV,SQI,SQV)
      RO=0.45;RI=0.65;NOCCUP=25;SPVENT=5.0
      CALL AIRPRO(TI,RI,HIAIR,SVR,W)
      CALL AIRPRO(TO,RO,HOAIR,SVO,W)
      CALL LATENT(TI,TO,RH,HW)
      CALL AIRCNG(VROOM,ANOCNG)
      QI=VROOM*ANOCNG*(HOAIR-HIAIR)/(SVO*24*3600)
      QV=NOCCUP*SPVENT*(HOAIR-HIAIR)/(SVO*24*3600)
      SQI=QI-(VROOM*ANOCNG*(HOAIR-HW)/(SVO*24*3600))
      SQV=QV-(NOCCUP*SPVENT*(HOAIR-HW)/(SVO*24*3600))
      RETURN;END
*----------------------------------------------------------------------------*
*           SUBROUTINE FOR AIR PROPERTIES                                    *
*----------------------------------------------------------------------------*
      SUBROUTINE AIRPRO(TDB,RH,H,SPVOL,WS)
      CPA=1.004;RA=287.2
      X=1.152E-5-(4.787E-9)*(TDB+273.15)
      Y=(7.21379+(X*(TDB-210.0)**2.0))*(647.31/(TDB+273.15)-1.0)
      PS=(221.228/EXP(Y))
      FS=1.004505-(2.0707E-5)*TDB+(9.145E-7)*(TDB)**2.0
      WS=0.622*FS*PS/(1.013952-FS*PS)
      H=1.004*TDB+WS*(2501.4+1.88*TDB)
      SPVOL=RA*(TDB+273.15)/((1.013952-RH*PS)*1E+5)
```

```
      RETURN;END
*-----------------------------------------------------------------------*
*          SUBROUTINE FOR NUMBER OF AIR CHANGES                         *
*-----------------------------------------------------------------------*
      SUBROUTINE AIRCNG(VROOM,ANOCNG)
      DIMENSION VOL(31),AIRCNG(31)
      DATA(AIRCNG(I),I=1,31)/22.5,20.5,19.25,18.25,17.37,16.6,15.91,
     115.29,14.74,14.25,13.80,13.4,13.03,12.7,12.4,12.12,11.87,11.67,
     111.43,10.26,9.22,8.44,8.13,7.85,7.37,6.99,6.42,6.3,6.01,5.8,
     15.55/
      DATA(VOL(I),I=1,31)/5,10,15,20,25,30,35,40,45,50,55,60,65,70,75,
     180,85,90,95,100,120,140,150,160,180,200,240,250,280,300,350/
      I=0.0
    1 I=I+1
      IF(VOL(I)-VROOM)1,2,2
    2 DIFF=VROOM-VOL(I)
      ANOCNG=AIRCNG(I)+DIFF*(AIRCNG(I-1)-AIRCNG(I))/(VOL(I)-VOL(I-1))
      RETURN;END
*-----------------------------------------------------------------------*
*          SUBROUTINE FOR LATENT HEAT                                   *
*-----------------------------------------------------------------------*
      SUBROUTINE LATENT(TI,TO,RH,HW)
      CALL AIRPRO(TI,RH,H,SPVOL,W)
      HW=1.004*TO+W*(2501.4+1.88*TO)
      RETURN;END
*&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&&*
```

```
***************************************************************************
*     COMPUTER PROGRAM FOR NON-LINEAR PROGRAMMING OPTIMISATION            *
*               This NLP optimisation is done                             *
*                             by                                          *
*               "INTERIOR PENALTY FUNCTION METHOD"                        *
*                  with INEQUALITY CONSTRAINTS                            *
*              Unconstrained minimisation is done by                      *
*               "DAVIDON-FLETCHER-POWELL'S METHOD"                        *
*              One dimensional minimisation is done by                    *
*                "CUBIC INTERPOLATION TECHNIQUE"                          *
*------------------------- SOURCE LISTING --------------------------------*
*      N     :Number of design variables.                                 *
*      NC    :Number of constraints.                                      *
*      NR    :Number of penalty parameter reductions desired.             *
*      ITLIM:Maximum number of D.F.P. iterations required.                *
*     LIMFIT:Maximum number of cubic refits.                              *
*      RI    :Initial value for the penalty parameter.                    *
*      Z     :Multiplication factor for penalty parameter(RI).            *
*      RF    :Reduction factor for penalty parameter(RO).                 *
*      EPSS  :Accuracy for the optimum design variables.                  *
*      EPC   :Accuracy for the gradients .                                *
*      ZZ    :Initial value for the step length.                          *
*      XX    :Factor for the step length to achieve convergence.          *
*   (X(I),I=1,N) : Starting values for the design variables.              *
*  (EPSM(I),I=1,N):Accuracy parameters uses in D.F.P. iterations.         *
***************************************************************************
!............................ MAIN PROGRAM...............................!
--------------------------------------------------------------------------
      DIMENSION X(20),GGI(10),EPSM(5)
      COMMON/BLOCK1/EPSS,R,ITER,INI,N,NC,LIMFIT,ZZ,XX,EPC
      OPEN(UNIT=1,FILE='FOR01.DAT',DEVICE='DSK',ACCESS='SEQIN')
      OPEN(UNIT=24,FILE='FOR24.DAT',DEVICE='DSK',ACCESS='SEQIN')
      OPEN(UNIT=25,FILE='MOD.DAT',DEVICE='DSK',ACCESS='SEQOUT')
      READ(1,*)Z,RF,TI,N,NC,NR,ITLIM,LIMFIT,RI,EPSS,(X(I),EPSM(I),
     1I=1,N),ZZ,XX,EPC
      INI=1
      NSTART=90
      II=0
      WRITE(25,1)TI
1     FORMAT(/,30X,'OPTIMISATION RESULTS FOR INSIDE TEMPERATURE:',F4.1
     1,/,50X,'SYSTEM (WITHOUT DESERT COOLER & WITH R22 REFRIGERANT)'
     2,/,30X,72('*'))
      WRITE(25,2)N,NC,NR,ITLIM,LIMFIT
2     FORMAT(//,30X,'OPTIMISATION PARAMETERS:',/,30X,24('-'),//,50X,
     1 'NO.OF DESIGN VARIABLES:',I3,/,50X,'NO.OF CONSTRAINTS:',I3,/
     1 ,50X,'NO.OF REDUCTIONS FOR THE PENALTY PARAMETER:',I3,/,
     1 50X,'MAX.NO. OF ITERATIONS FOR D.F.P.:',I3,/,50X,'MAX.NO.OF
     1 CUBIC REFITS:',I3)
*.....................Starting value of penalty parameter.................
      CALL PENAL(X,NC,N,RI,OBJ1,GG1,FB1,SUM1)
      RO=-(Z*OBJ1/SUM1)
      WRITE(25,3)RO
3     FORMAT(//,30X,'STARTING VALUE FOR THE PENALTY PARAMETER (RO):',
     1 E16.8,/,30X,45('-'),//,30X,'STARTING DESIGN VECTOR X(I):',/,
     1 30X,28('-'))
      WRITE(25,4)(X(I),I=1,N)
4     FORMAT(34X,E16.8,4X,E16.8,4X,E16.8,4X,E16.8,4X,E16.8)
*.........Beginning of "INTERIOR PENALTY FUNCTION METHOD"..............
      DO 5 IR=1,NR
      II=II+1
      IF(II.GT.5) II=5
      R=RO
      CALL OPTIM(X,II,EPSM,ITLIM,NSTART,F,OBJ)
      RO=RF*RO
```

```
5     CONTINUE
      STOP
      END
*****************************************************************************
*...........................SUBROUTINES........................................
*-----------------------------------------------------------------------------
*           SUBROUTINE FOR D.F.P. METHOD                                       *
*-----------------------------------------------------------------------------*
      SUBROUTINE OPTIM(X,II,EPSM,ITLIM,NSTART,F,OBJ)
      REAL NO,MO
      INTEGER C,V
      DIMENSION G(10),GT(10),X(20),S(20),GRAD(20),XB(20),XO(20),
     1HY(20),NO(20,20),MO(20,20),H(20,20),EPSM(5),GRADP(20),SN(20)
      DIMENSION GRADS(10)
      COMMON/BLOCK1/EPSS,R,ITER,INT,V,C,LIMFIT,ZZ,XX,EPO
      TYPE1
1     FORMAT(10X,'ENTERS DFP')
      WRITE(25,2)R
2     FORMAT(///,15X,'VALUE FOR PENALTY PARAMETER (R):',E16.8,/,15X,
     1 32('-'))
      WRITE(25,3)
3     FORMAT(80('*'),/,1X,'ITER.NO',2X,'PENALTY FUNC',3X,'OBJ.FUNC',
     1/,80('*'))
4     FORMAT(1X,'STARTING POINT  VIOLATES A CONSTRAINT.
     1X(I) ARE',/(1X,10E13.6))
      ITER=0
      INIT=1
      IF(II.GE.5) GO TO 5
      IF(II.EQ.1) T=ZZ
      INIT=0
      GO TO 18
5     CALL PENAL(X,C,V,R,OBJ,G,F,SUM2)
      INIT=1
      DO 6 I=1,C
      IF(G(I).GE.0.0) GO TO 55
6     CONTINUE
      CALL GRADN(X,V,C,R,G,F,GRAD)
      TYPE*,F,OBJ
      WRITE(25,7)(ITER,F,OBJ)
7     FORMAT(5X,I3,5X,F8.2,5X,F8.2)
      WRITE(25,8)(X(I),I=1,V)
8     FORMAT(2X,'DESIGN VECTORS:',5(2X,F7.4))
      IF(F.GT.OBJ)GO TO 17
      STOP
9     WRITE(25,10)
10    FORMAT(10X,'ABS(F-OBJ) < 2.0')
11    WRITE(25,12)
12    FORMAT(2X,'FX(I+1) AND FX(I) ARE < EPSILON ')
      WRITE(25,13)(ITER,F,OBJ)
13    FORMAT(5X,I3,5X,F8.2,5X,F8.2)
      WRITE(25,14)(X(I),I=1,V)
14    FORMAT(2X,'DESIGN VECTORS:',5(2X,F7.4))
      STOP
15    WRITE(25,16)
16    FORMAT(2X,'G(I) > 0.0')
17    GO TO 22
*...........Initialisation of unit matrix.......................................
18    DO 20 I=1,V
      DO 19 J=1,V
19    H(I,J)=0.0
20    CONTINUE
      DO 21 I=1,V
21    H(I,I)=1.0
      ITER2=0
```

```
      IF(INIT.EQ.0) GO TO 5
*..............Calculation of direction vector.......................
22    DO 24 I=1,V
      A=0.0
      DO 23 J=1,V
23    A=A+H(I,J)*GRAD(J)
24    S(I)=-A
*..............Normalisation of direction vectors....................
      SNL=0.0
      DO 25 I=1,V
25    SNL=(SNL+S(I)**2.0)
      SNL=SQRT(SNL)
      DO 26 I=1,V
26    SN(I)=S(I)/ABS(SNL)
      DO 27 I=1,V
27    GRADP(I)=GRAD(I)
      ITER=ITER+1
*..............Determination of optimal step length..................
      TYPE 28,ITER
28    FORMAT(5X,I2,5X,'ENTERS CUBE')
      IFLAG=0
      CALL INPOL(X,G,F,OBJ,SN,GRAD,IFLAG,T)
      DO 29 I=1,V
29    XB(I)=X(I)-T*SN(I)
      TYPE 30
30    FORMAT(10X,'COMES OUT OF CUBE')
      SUM=0.0
      DO 31 I=1,V
31    SUM=SUM+GRAD(I)**2
      SUM=SQRT(SUM)
      IF(SUM.LT.EPSS)GO TO 49
      SUM=0.0
      NN=1
32    DO 33 I=1,C
      IF(G(I).GT.0.0) T=.9*T
      IF(G(I).GT.0.0) GO TO 36
33    CONTINUE
      TYPE 34
34    FORMAT(15X,'G(I)<0.0')
      DO 35 I=1,V
      IF(ABS(GRAD(I)).LE.1.0)GO TO 39
35    CONTINUE
      GO TO 40
36    DO 37  I=1,V
37    X(I)=XB(I)+T*SN(I)
      TYPE 38
38    FORMAT(5X,'G(I)>0.0')
      CALL PENAL(X,C,V,R,OBJ,G,F,SUM3)
      SUM=SUM+NN
      IF(SUM.GT.25.0)GO TO 15
      GO TO 32
39    WRITE(20,52)(GRAD(I),I=1,V)
40    WRITE(25,7)ITER,F,OBJ
      WRITE(25,8)(X(I),I=1,V)
      IF(F.LT.OBJ)GO TO 53
      TYPE 41
41    FORMAT(10X,'F >OBJ')
      IF(ABS(F-OBJ).LE.2.0)GO TO 9
      IF(ITER.GT.ITLIM) GO TO 49
      ITER2=ITER2+1
      IF(ITER2.EQ.NSTART) GO TO 18
      DO 42 I=1,V
42    YQ(I)=GRAD(I)-GRADP(I)
*......................Updating of the unit matrix....................
```

```
      STY=0.0
      DO 43 I=1,V
43    STY=STY+(SN(I)*YQ(I))
      STY=STY/T
      DO 45 I=1,V
      A=0.0
      DO 44 J=1,V
44    A=A+H(I,J)*YQ(J)
45    HY(I)=A
      YHY=0.0
      DO 46 I=1,V
46    YHY=YHY+HY(I)*YQ(I)
      DO 48 I=1,V
      DO 47 J=1,V
      MQ(I,J)=(SN(I)*SN(J))/STY
      NQ(I,J)=-(HY(I)*HY(J))/YHY
47    H(I,J)=H(I,J)+MQ(I,J)+NQ(I,J)
48    CONTINUE
      GO TO 22
49    CONTINUE
      WRITE(25,50)
50    FORMAT(//,20X,'VALUES FOR THE CONSTRAINTS:',/,20X,47('-'))
      WRITE(25,51)(G(I),I=1,C)
51    FORMAT(20X,E16.8,5X,E16.8,5X,E16.8,5X,E16.8,5X,E16.8)
52    FORMAT(4X,'GRADIENTS ARE:',/,4X,5(2X,E13.6))
      RETURN
53    WRITE(25,54)
54    FORMAT(10X,'FUNCTION VALUE = OBJ.VALUE')
55    WRITE(25,4),(X(I),I=1,V)
      WRITE(25,51)(G(I),I=1,C)
      RETURN
      END
*-------------------------------------------------------------------------*
*             SUBROUTINE FOR PENALTY FUNCTION.                            *
*-------------------------------------------------------------------------*
      SUBROUTINE PENAL(X,NC,NDV,RK,OBJ,GG,FB,SUM)
      DIMENSION GG(10),X(20)
      CALL CONS(X,GG)
      CALL OBJT(X,GG,OBJ)
*......................Calculation of the penalty function...............
      SUM=0.0
      DO 1 I=1,NC
      IF(GG(I).GE.0.0) GO TO 1
      SUM=SUM+1./GG(I)
 1    CONTINUE
      FB=OBJ-RK*SUM
      RETURN
      END
*-------------------------------------------------------------------------*
*          SUBROUTINE FOR THE OBJECTIVE FUNCTION EVALUATION.              *
*-------------------------------------------------------------------------*
!                              the objective function is the total cost   !
!Here, in this program ,                                                  !
!                                     vapour compression system per year. !
!        involved in the                                                  !
!                Total cost=Crun+Cini                                     !
!    where Crun =(Running cost+Maintenance cost)/year (in Rupees)         !
!                Cini: initial cost for equipments/year    (in Rupees)    !
!------------------------------DESIGN VARIABLES---------------------------!
!     X(1) :  Degree of rise in higher temperature (deg.Celcius)          !
!     X(2) :  Degree of fall in lower temperature  (deg.Celcius)          !
!     X(3) :  Degree of superheat                  (deg.Celcius)          !
!     X(4) :  Degree of subcooling                                        !
!*************************************************************************!
!     SOURCE LISTING:                                                     !
!     TROOM,RHROOM  : Temperature & Relative humidity of the air          !
```

```
                      inside the room.
      TOUT,ROUT     : Temperature & Relative humidity of the air
                      outside the room.
COMEFF,YETAF,YETAM  : Efficiencies of the compressor,blower fan &
                      Mechanical respectively.
      NMON,NRUN     : Number of months & hours in which the system
                      runs.
    CDEST,RATDES    : Cost & Rating of the Desert Cooler.
   EVTHIC,COTHIC    : Wall thickness of the Condenser & Evaporator
      LIFE,IRATE    : Life & Rate of interest for the system.
              SF    : Factor of safety for the system.
           JN,KN    : Reference index.
                    : JN=1 (system working with R12 refrigerant)
                    : JN=2 (system working with R22 refrigerant)
                    : KN=1 (system working without Desert Cooler)
                    : KN=2 (system working with Desert Cooler)
           DELTA    : Approach temperature in the Condenser &
                      Evaporator.
          NOCCUP    : Number of occupants inside the room.
         TO1(I,J)   : Outside air temperature variation for the
                      running hours of the system.
         RH1(I,J)   : Relative humidity of the outside air
      SENVE1(I,J)   : Ventilation load during the running hours of
                      the system.
      SLOAD1(I,J)   : Sensible heat load.
      TLOAD1(I,J)   : Total cooling load.
*****************************************************************
      SUBROUTINE OBJT(X,GG,CTOT)
      REAL MREF,IRATE,MAIR
      DIMENSION X(10),GG(10)
      DIMENSION TO1(5,10),RH1(5,10),SLOAD1(6,10),SENVE1(6,10),TLOAD1
     1(6,10),COMCOS(5),CM1COS(5),COMCAP(10),CM1CAP(10),
     2EVACOS(5),EVICOS(5),EVACAP(5),EVICAP(5),CONCOS(5),CN1COS(5),
     3CONCAP(5),CN1CAP(5),BLOCAP(5),BLOCOS(5)
      CPA=1.004
      RHO=1.2
 1    FORMAT(4X,'I=',I3,5X,'X(I)=',F7.3)
      READ(24,*)TROOM,RHROOM,COMEFF,SF,YETAF,LIFE,IRATE,EVTHIC,
     1TOUT,RHOUT,NRUN,JN,YETAM,DELTA,VP,NOCCUP,COTHIC,NMON
     2,KN,Y,M,CDEST,RATDES
      DO 3 I=1,NMON
      DO 2 J=1,NRUN
      READ(24,*)TO1(I,J),RH1(I,J),SENVE1(I,J),SLOAD1(I,J),TLOAD1(I,J)
 2    CONTINUE
 3    CONTINUE
      ENDFILE 24
      REWIND 24
C.................. Calculation of initial cost..................
      BIG=0.0
      I=2
      IF(KN.NE.1)I=4
      DO 5 J=1,NRUN
      IF(BIG-TLOAD1(I,J)) 4,5,5
 4    BIG=TLOAD1(I,J)
      KOUNT=J
 5    CONTINUE
      K=KOUNT
      SLOAD=SLOAD1(I,K)
      QLOAD=TLOAD1(I,K)
      SENVEN=SENVE1(I,K)
      TO=TO1(I,K)
      RH=RH1(I,K)
      TH=TO+DELTA+X(1)
      TL=TROOM-DELTA-X(2)
```

```
      T1=TL+X(3)
      SUBT=TH-X(4)
      IF(JN.EQ.2)GO TO 6
      CALL R12VAL(TH,TL,T1,SUBT,COMEFF,H1,H2,H3)
      GO TO 7
 6    CALL R22VAL(TH,TL,T1,SUBT,COMEFF,H1,H2,H3)
 7    QLOAD=SF*QLOAD
      MREF=QLOAD/(H1-H3)
      PCOMP1=MREF*(H2-H1)/(3.5*YETAM)
      QCOND=MREF*(H2-H3)/3.5
      EVAPR=QLOAD/3.5
      MAIR=QCOND*3.5/(CPA*X(1))
      VAIR=(MAIR*60.0)/RHO
      CALL COSTIN(JN,PCOMP1,QCOND,EVAPR,VAIR,CTOTAL)
      IF(KN.EQ.1)GO TO 8
      CTOTAL=CTOTAL+CDEST
      CALL ELEC(IRATE,LIFE,CE)
      RUNDES=RATDES*CE*90
      M=4
      GO TO 9
*......................Calculation of running cost per year.................
 8    RUNDES=0.0
      CALL ELEC(IRATE,LIFE,CE)
 9    CONTINUE
      CR=0.0
      CALL IR(IRATE,LIFE,C5)
      DO 13 I=M,NMON
      RUN=CR
      CRUN=0.0
      DO 12 J=1,NRUN
      TH=TO1(I,J)+DELTA+X(1)
      TL=TROOM-DELTA-X(2)
      T1=TL+X(3)
      SUBT=TH-X(4)
      IF(JN.EQ.2)GO TO 10
      CALL R12VAL(TH,TL,T1,SUBT,COMEFF,H1,H2,H3)
      GO TO 11
 10   CALL R22VAL(TH,TL,T1,SUBT,COMEFF,H1,H2,H3)
 11   MREF=SF*TLOAD1(I,J)/(H1-H3))
      PCOMP=MREF*(H2-H1)/YETAM
      MAIR=(SLOAD1(I,J)+BF*SENVEL(I,J))/(CPA*(TO1(I,J)-TROOM))
      MAIR=MAIR*1.0
      CMAIR=(MREF*(H2-H3))/(CPA*X(1))
      VOLAIR=CMAIR/RHO
      EVAIR=MAIR/RHO
      PBLOW=(3.429*VOLAIR*VOLAIR+1.443*VOLAIR+37.67)/(1000*YETAF)
      FPOWER=(3.429*EVAIR*EVAIR+1.443*EVAIR+37.67)/(1000*YETAF)
      C1=PCOMP*CE
      C2=FPOWER*CE
      C3=PBLOW*CE
      CRUN=CRUN+C1+C2+C3
 12   CONTINUE
      CR=CRUN*30.0+RUN
 13   CONTINUE
*......................Calculation of total cost.............................
      CINI=CTOTAL*(1.2)/LIFE
      CRUN=CR+0.4*CINI*C5+RUNDES
      CTOT=(CRUN+CINI)/CE
      RETURN
      END
*-------------------------------------------------------------------------*
*          SUBROUTINE FOR THE CONSTRAINTS.                                *
*-------------------------------------------------------------------------*
      SUBROUTINE CONS(X,GG)
```

```
      DIMENSION GG(10),X(20)
      GG(1)=1.0-X(1)/4.0
      GG(2)=X(1)/15.0-1.0
      GG(3)=1.0-X(2)/2.0
      GG(4)=X(2)/5.0-1.0
      GG(5)=1.0-X(3)
      GG(6)=X(3)/4.0-1.0
      GG(7)=1.0-X(4)
      GG(8)=X(4)/4.0-1.0
      RETURN
      END
*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*
*               SUBROUTINE FOR CUBIC INTERPOLATION                               *
*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*-*
      SUBROUTINE INPOL(X,G,FC,OBJ,S,GRAD,IFLAG,T)
      INTEGER C,V,QQ
      REAL LTHS,LTHG
      DIMENSION X(20),S(20),GRAD(20),XT(20),GRADT(20),GT(90),G(90)
      COMMON/BLOCK1/EPSS,R,ITER,INI,V,C,LIMFIT,ZZ,XX,EPC
    1 FORMAT(2X,6(I4,2X,E15.7))
    2 FORMAT(2X,37H POSSIBILITY OF LOCAL MINIMA DETECTED )
    3 FORMAT(2X,14H KLMT2 GTR 25 )
    4 FORMAT(2X,14H  KLMT3 GTR 35)
    5 FORMAT(2X,14H  KLMT1 GTR 50)
    6 FORMAT(2X,I5,8E15.7/(5X,8E15.7))
      LOCMIN=0
      IPR2=0
      IPR=0
      IF(IFLAG.NE.0) IPR2=1
      IF(IFLAG.NE.0) IPR=1
      IFLAG=0
      IF(IPR.NE.0) WRITE(22,6),ITER,(S(I),I=1,V)
      IF(IPR.NE.0) WRITE(22,6),ITER,(G(I),I=1,C)
      KLMT1=0
      KLMT2=0
      KLMT3=0
      QQ=1
    7 D=1.0
      FA=FC
      FAP=0.0
      DO 8 J=1,V
    8 FAP=FAP+S(J)*GRAD(J)
      IF(FAP.GE.0.0) GO TO 38
      LTHS=0.0
      DO 9 J=1,V
    9 LTHS=LTHS+S(J)*S(J)
      LTHS=SQRT(LTHS)
      T0=ABS(T)
      IF(T0.LT.0.0001) T0=0.0001
      T=T0
      TINC=T
      A=0.0
   10 CONTINUE
      IDENT=1
      IF(IPR.NE.0) WRITE(22,6),IDENT,A,FA,FAP,T,TINC
      DO 11 J=1,V
   11 XT(J)=X(J)+T*S(J)
      IF(INI.GT.V) GO TO 14
      DO 12 J=INI,V
      IF(XT(J).LT.0.0) GO TO 13
   12 CONTINUE
      GO TO 14
   13 CONTINUE
      TINC=(T-A)*0.6
```

```
      T=T-TINC
      KLMT2=0
      KLMT3=KLMT3+1
      KLMT1=KLMT1+1
      IF(KLMT1.GE.75) GO TO 34
      GO TO 10
14    CONTINUE
      CALL PENAL(XT,C,V,R,OBJ,GT,FB,SUMT)
      IDENT=2
      IF(IPR.NE.0) WRITE(22,6),IDENT,(G(I),I=1,C)
      DO 15 I=1,C
      IF(G(I).GT.0.0) GO TO 15
      IF(GT(I).GT.0.0) GO TO 13
15    CONTINUE
      KLMT1=0
      KLMT3=KLMT3+1
      KLMT2=KLMT2+1
      CALL SLOPE(XT,GT,S,GRADT,FB,FBP)
      IDENT=3
      IF(IPR.NE.0) WRITE(22,6),IDENT,FB,FBP,FA,FAP
      IF(FBP.GT.0.0) GO TO 21
      IF(KLMT2.GE.50) GO TO 36
      IF(KLMT3.GE.75) GO TO 35
      IF(FB.GT.FA) GO TO 38
      GO TO 20
16    CONTINUE
      DO 18 J=1,V
17    GRAD(J)=GRADT(J)
18    X(J)=XT(J)
      IF(IPR2.NE.0.0) WRITE(22,6),ITER,(GRAD(J),J=1,V)
      IF(IPR2.NE.0.0) WRITE(22,6),ITER,(X(J),J=1,V)
      DO 19 I=1,C
19    G(I)=GT(I)
      RETURN
20    CONTINUE
      FAP=FBP
      FA=FB
      A=T
      T=XX*T
      GO TO 10
21    CONTINUE
      B=T
22    Z=(3.0*(FA-FB))/(B-A)+FAP+FBP
      TSTA=ABS(FAP)+ABS(FBP)+2.0*ABS(Z)
      TSTB=FAP+FBP+2.0*Z
      IF(ABS(TSTB/TSTA).LT.1.0E-07) GO TO 39
      TSTA=1.0-(FAP/Z)*(FBP/Z)
      QZ=ABS(Z)*SQRT(TSTA)
      T=A+((FAP+Z+QZ)*(B-A))/TSTB
23    CONTINUE
24    DO 25 J=1,V
25    XT(J)=X(J)+T*S(J)
      CALL PENAL(XT,C,V,R,OBJ,GT,D,SUMT1)
      DO 26 IJK =1,C
      IF(GT(IJK).GE.0.0) T=0.9*T
      IF(GT(IJK).GE.0.0) GO TO 24
26    CONTINUE
      IDENT=4
      IF(IPR.NE.0) WRITE(22,6),IDENT,Z,QZ,T,D,OBJ
      DO 27 I=1,C
      IF(G(I).GE.0.0) GO TO 27
      IF(GT(I).GE.0.0) GO TO 33
27    CONTINUE
      CALL SLOPE(XT,GT,S,GRADT,D,STG)
```

```
      LTHG=0.0
      DO 28 I=1,V
   28 LTHG=LTHG+GRADT(I)**2
      LTHG=SQRT(LTHG)
      TSTA=STG/(LTHS*LTHG)
      IF(IPR2.NE.0) WRITE(22,29) ,TSTA,STG,LTHS,LTHG
   29 FORMAT(11H QUOTIENT =,E12.5,3X,7HSLOPE =,E12.5,2X,6HLTHS =,
     1E12.5,2X,6HLTHG =,E12.5)
      IF(ABS(TSTA).GT.EPSS) GO TO 34
      QQ=QQ+1
      IF(QQ.GT.LIMFIT) GO TO 31
      IF(ABS(STG).LE.EPCD)GO TO 30
      A=T
      FA=D
      FAP=STG
      GO TO 22
   30 B=T
      FB=D
      FBP=STG
      GO TO 22
   31 CONTINUE
   32 FB=D
      GO TO 16
   33 IF(LOCMIN.EQ.0) WRITE(20,2)
      LOCMIN=1
      GO TO 13
   34 WRITE(20,5)
      STOP
   35 WRITE(20,4)
      GO TO 37
   36 WRITE(20,3)
   37 WRITE(20,1),(J,XT(J),J=1,V)
      WRITE(20,1),(J,GT(J),J=1,CD)
      IFLAG=1
      GO TO 16
   38 WRITE(22,*)FAP,ITER
      STOP
   39 T=A+(B-A)*FAP/(2.0*(Z+FAP))
      QZ=0.0
      GO TO 23
      END
*-----------------------------------------------------------------------*
*                SUBROUTINE FOR GRADIENTS.                              *
*-----------------------------------------------------------------------*
      SUBROUTINE GRADN(X,N,NC,R,G,F,GRAD)
      DIMENSION X(20),G(10),GRAD(20),XP(20),XM(20)
      DO 1 I=1,N
      XP(I)=X(I)
    1 XM(I)=X(I)
      DO 2 I=1,N
      XP(I)=1.01*X(I)
      XM(I)=0.99*X(I)
      CALL PENAL(XP,NC,N,R,OBJ,G,FBP,SUMP)
      CALL PENAL(XM,NC,N,R,OBJ,G,FBM,SUMM)
      XP(I)=X(I)
      XM(I)=X(I)
    2 GRAD(I)=(FBP-FBM)/(0.02*X(I))
      RETURN
      END
*-----------------------------------------------------------------------*
*                SUBROUTINE FOR SLOPE.                                  *
*-----------------------------------------------------------------------*
      SUBROUTINE SLOPE(X,G,S,GR,F,FBP)
      DIMENSION G(10),X(20),S(20),GR(20)
```

```
      INTEGER C,V
      COMMON/BLOCK1/EPSS,R,ITER,INT,V,C,LIMFIT
      CALL GRADN(X,V,C,R,G,F,GR)
      FBP=0.0
      DO 1 I=1,V
    1 FBP=FBP+S(I)*GR(I)
      RETURN
      END
*-----------------------------------------------------------------------*
*                 SUBROUTINE FOR AIR PROPERTIES.                        *
*-----------------------------------------------------------------------*
      SUBROUTINE AIRPRI(TDB,RH,PS,SPVOL,WS,W,H)
      CPA=1.004
      RA=287.2
      X=1.152E-5-(4.787E-9)*(TDB+273.15)
      Y=(7.21379+(X*(TDB-210.0)*(TDB-210.0)))*(647.31/(TDB+273.15)-1.0)
      PS=(221.228/EXP(Y))
      SPVOL=RA*(TDB+273.15)/((1.013962-RH*PS)*1E+5)
      FS=1.004505-(2.0707E-5)*TDB+(9.145E-7)*(TDB)**2
      WS=0.622*FS*PS/(1.013962-FS*PS)
      H=1.004*TDB+WS*(2501.4+1.88*TDB)
      Z=FS*PS/1.013962
      DEGSAT=RH*(1.0-Z)/(1.0-RH*Z)
      W=DEGSAT*WS
      RETURN
      END
*-----------------------------------------------------------------------*
      SUBROUTINE ELEC(R,L,COST)
      DIMENSION CE(20)
      DO 1 I=1,20
      CE(I)=1.293/(0.97985+EXP(-0.09388*I))
    1 CONTINUE
      AL=L
      CEFF=0.0
      DO 2 I=1,L
      CEFF=CEFF+CE(I)/(1.0+R)**(I-1)
    2 CONTINUE
      COST=CEFF/AL
      RETURN
      END
*-----------------------------------------------------------------------*
      SUBROUTINE IR(R,L,C5)
      X1=(1.0+R)**L-1.0
      X2=R*(1.0+R)**(L-1)
      C5=X1/X2
      RETURN
      END
*-----------------------------------------------------------------------*
*     CAPACITY AND COST OF R12 & R22 EQUIPMENTS                         *
*-----------------------------------------------------------------------*
      SUBROUTINE COSTIN(J,PCOMP1,QCOND,EVAPR,VAIR,CCOST)
      DIMENSION COMCAP(5),CONCAP(5),EVACAP(5),COMCOS(5),CONCOS(5)
     1,EVACOS(5),BLOCOS(5),BLOCAP(5)
      DIMENSION CM1CAP(5),CN1CAP(5),EV1CAP(5),CM1COS(5),CN1COS(5),
     1EV1COS(5)
      IF(J.EQ.2)GO TO 9
      DATA(COMCAP(I),I=1,5)/0.5,1.0,1.5,2.0,3.0/
      DATA(COMCOS(I),I=1,5)/4000.0,4800.0,5500.0,8000.0,10000.0/
      DATA(CONCAP(I),I=1,5)/0.5,1.0,1.5,3.0,4.0/
      DATA(CONCOS(I),I=1,5)/1800.0,3500.0,4200.0,4800.0,5500.0/
      DATA(EVACAP(I),I=1,5)/0.5,1.0,2.0,3.0,4.0/
      DATA(EVACOS(I),I=1,5)/2200.0,3500.0,4200.0,4900.0,5800.0/
      DATA(BLOCAP(I),I=1,5)/50.0,80.0,120.0,150.0,200.0/
      DATA(BLOCOS(I),I=1,5)/1100.0,1700.0,2000.0,2500.0,3100.0/
```

```
      I=0
1     I=I+1
      IF(PCOMP1-COMCAP(I))2,2,1
2     PCOMP=COMCAP(I)
      CPCOST=COMCOS(I)
      I=0
3     I=I+1
      IF(QCOND-CONCAP(I))4,4,3
4     QCOND=CONCAP(I)
      CTOTA=CONCOS(I)+CPCOST
      I=0
5     I=I+1
      IF(EVAPR-EVACAP(I))6,6,5
6     EVAPR=EVACAP(I)
      CEQP=EVACOS(I)+CTOTA
      I=0
7     I=I+1
      IF(VAIR-BLOCAP(I))8,8,7
8     VAIR=BLOCAP(I)
      CCOST=BLOCOS(I)+CEQP
      RETURN
9     CONTINUE
      DATA(CM1CAP(I),I=1,5)/0.5,1.0,1.5,2.0,3.0/
      DATA(CM1COS(I),I=1,5)/5000.0,6000.0,7000.0,85000.0,11000.0/
      DATA(CN1CAP(I),I=1,5)/0.5,1.0,1.5,2.0,3.0/
      DATA(CN1COS(I),I=1,5)/2200.0,4000.0,5500.0,7000.0,8000.0/
      DATA(EV1CAP(I),I=1,5)/0.5,1.0,1.5,2.0,3.0/
      DATA(EV1COS(I),I=1,5)/2100.0,3600.0,4600.0,5800.0,6900.0/
      DATA(BLOCAP(I),I=1,5)/50.0,30.0,120.0,150.0,200.0/
      DATA(BLOCOS(I),I=1,5)/1100.0,1700.0,2000.0,2500.0,3100.0/
      I=0
10    I=I+1
      IF(PCOMP1-CM1CAP(I))11,11,10
11    PCOMP=CM1CAP(I)
      CPCOST=CM1COS(I)
      I=0
12    I=I+1
      IF(QCOND-CN1CAP(I))13,13,12
13    QCOND=CN1CAP(I)
      CTOTA=CN1COS(I)+CPCOST
      I=0
14    I=I+1
      IF(EVAPR-EV1CAP(I))15,15,14
15    EVAPR=EV1CAP(I)
      CEQP=EV1COS(I)+CTOTA
      I=0
16    I=I+1
      IF(VAIR-BLOCAP(I))17,17,16
17    VAIR=BLOCAP(I)
      CCOST=BLOCOS(I)+CEQP
      RETURN
      END
*-----------------------------------------------------------------------*
*     SUBROUTINE FOR PROPERTIES OF R12 REFRIGERANT                      *
*-----------------------------------------------------------------------*
      SUBROUTINE R12VAL(TH,TL,T1,SUBT,EFF,H1,H2,H3)
      H1=HG(TL)+CP(TL0*(T1-TL)
      S1=(HG(TL)-HF(TL))/(TL+273.15)+SF(TL)+CP(TL)*ALOG(
     1(T1+273.15)/(TL+273.15))
      TH=TH+4.0
      S2D=(HG(TH)-HF(TH))/(TH+273.15)+SF(TH)
      T21=(TH+273.15)*EXP((S1-S2D)/CP(TH))
      T2D=T21-273.15
      H2D=HG(TH)+CP(TH)*(T2D-TH)
```

```
      H2=(H2D-H1)/EFF+H1
      T100=TH-4.0
      H3=HF(SUBT)
      RETURN
      END
*.....................FUNCTION SUBROUTINES....................................
      FUNCTION HG(T)
      HG=188.86+0.440278*T/100.0*100.0-7.02007*(T/100.0)**2-5.07651
     1*(T/100.0)**3-3.82545*(T/100.0)**4
      RETURN
      END
      FUNCTION HF(T)
      HF=36.1554+0.928108*T/100.0*100.0+6.89916*T/100.0*T/100.0
     1+3.73414*(T/100.0)**3+5.91573*(T/100.0)**4
      RETURN
      END
      FUNCTION CP(T)
      CP=0.59524+0.0018175*(T+15)
      RETURN
      END
      FUNCTION SF(T)
      SF=0.142093+0.38747*T/100.0-0.0309606*T/100.0*T/100.0
     1+0.019869*(T/100.0)**3+0.014023*(T/100.0)**4
      RETURN
      END
*-----------------------------------------------------------------------------*
*     SUBROUTINE FOR PROPERTIES OF R22 REFRIGERANT                            *
*-----------------------------------------------------------------------------*
      SUBROUTINE R22VAL(TH,TL,T1,SUBT,EFF,H1,H2,H3)
      H1=HG1(TL)+CP1(TL)*(T1-TL)
      S1=(HG1(TL)-HF1(TL))/(TL+273.15)+SF1(TL)+CP1(TL)*ALOG(
     1(T1+273.15)/(TL+273.15))
      TH=TH+4.0
      S2D=(HG1(TH)-HF1(TH))/(TH+273.15)+SF1(TH)
      T21=(TH+273.15)*EXP((S1-S2D)/CP1(TH))
      T2D=T21-273.15
      H2D=HG1(TH)+CP1(TH)*(T2D-TH)
      H2=(H2D-H1)/EFF+H1
      T100=TH-4.0
      H3=HF1(SUBT)
      RETURN
      END
*................ FUNCTION SUBROUTINES.........................................
      FUNCTION HG1(T)
      HG1=251.06+(T/100.0)*(35.4577-(T/100.0)*(19.4993+(T/100.0)*
     1(7.62005+11.6756*(T/100.0))))
      RETURN
      END
      FUNCTION HF1(T)
      HF1=46.2101+T*1.20394+(T/100)*(T/100)*(6.828-(T/100)*(8.99337+
     1(T/100)*20.0612))
      RETURN
      END
      FUNCTION CP1(T)
      CP1=0.70114+0.0029529*(T-10.0+0.010472*((T-10.0)**2.0))
      RETURN
      END
      FUNCTION SF1(T)
      SF1=0.181132+(T/100)*(0.43657-(T/100)*(0.0596755+(T/100)*(0.0234
     1343-0.0638047*(T/100))))
      RETURN
      END
*SSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSSS*
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